JUN 27 1911

June 27, 1911.

PUBLIC LIBRARY
THE
CITY OF DETROIT

# POWER

Issued Weekly by the Hill Publishing Company, 505 Pearl St., New York

Price 5 Cents, $2.00 per Year

Contents 1009. Advertising Index 65


## Albany Grease

A thoroughly high grade lubricant for use on all moving parts of machinery. It is a solid lubricant; it is composed only of ingredients possessing the highest lubricating qualities; it will not waste, drip or gum.

And, consequently, to use Albany Grease is to reduce your lubricating bills, to insure cleanliness and lessen the cost of insurance, to minimize friction and to make your machinery run most easily and smoothly and with the lowest consumption of power. Albany Grease has been used far and wide for the past 43 years, a record that can mean but one thing—that Albany Grease has steadily given satisfaction wherever and for whatever used.

**Let us prove these superiorities to you. Send for FREE Albany Grease Cup and FREE sample of Albany Grease.**

**SIGN THE COUPON.**

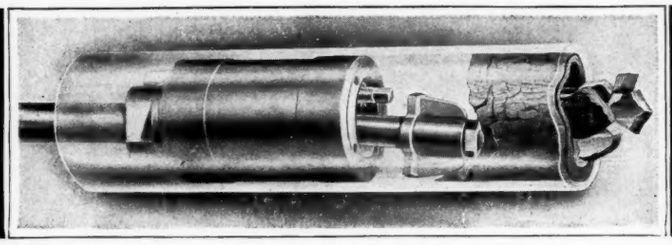
Albany Lubricating Co. (P), 708 Washington St., New York City. Send me, without obligation on my part, sufficient Albany Grease FREE for thirty days' trial and FREE Albany Grease Cup.

Name.....
Firm.....
Address.....
Pipe Tap.....
Depth of hole.....
Where used.....

Albany Lubricating Company

ADAM COOK'S SONS, Proprietors

708-710 Washington Street, New York City



The Dean Removing Scale from the Tube of a Water Tube Boiler.

# The Engineer's Opportunity

Getting scale off boiler tubes isn't the nicest job about the boiler room, by any means.

But it's about the best paying thing you can do.

Do the job as it ought to be done and

You can save coal, save the boilers, save steam and save labor.

You can save your company $150 to $1200 a year, and more!

That will make the Old Man sit up and take notice.

But you must go at the scale problem right.

Don't let *guess work* guide you.

Don't *guess* you have no scale.

Don't *guess* you can't get better results than you are now getting.

Because you may be mistaken.

Because an investigation means a gain more often than a loss.

For instance

# The Dean Boiler Tube Cleaner

has proved a wonder as a money saver, nine times out of ten.

Simply because it removes scale from either water tube or fire tube boilers better than anything else.

Yet it required an investigation to find this out.

Now we offer you an opportunity to find out whether your boilers are clean, whether your compounds are doing what you think they are doing, whether you are saving all that it is possible to save.

We offer to loan you a Dean for free trial in one of your boilers so you can answer these questions.

Clean the boiler by the Dean way; then watch your coal pile and your steaming.

Compare old results with Dean results and you won't want to be without a Dean at six times the price we ask for it.

Then, but not till then, we'll talk over the matter of purchase.

And if you purchase, we'll guarantee the Dean to pay for itself in six months; otherwise you can return it and get your money back.

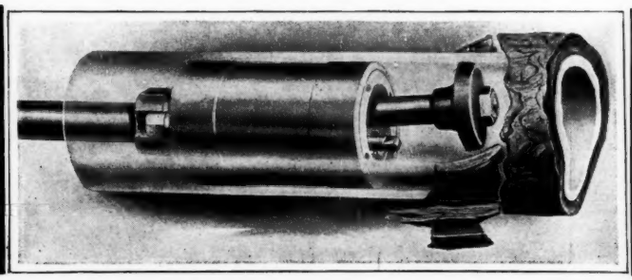
Shall we send you a Dean on trial?

"Scale Removal Made Easy" is worth dollars and cents to the engineer—for the asking you can get a copy free.

## The Wm. B. Pierce Company

**Jewett Building, Buffalo, N. Y.**

**Chicago Office: 801 Steinway Bldg.**



The Dean Removing Scale from the Tube of a Return Tubular Boiler.

# POWER

Vol. 33 NEW YORK, JUNE 27, 1911 No. 26

OBSERVE the surprised and agonized look on the face of Alvah, Jr.

Why is Al so astonished and pained?

Simply because he has just eaten one green little apple; that's all. The pained expression is the result of the colic the apple has given him; his surprise is due to the fact that he *didn't know* that green apples cause gripes.

* * *

Every little while you read about someone accidentally shooting a sister, brother or dear friend. The fool always pleads: "I *didn't know* it was loaded."

* * *

Continually, gay youth is rocking the boat. When, as a result, two or three are drowned, the idiot whimpers that he *didn't know* he was incurring any danger.

* * *

Alvah's lack of knowledge concerning the painful possibilities of green apples does not mitigate the discomfort he is suffering amidships.

Because the unintentional murderer didn't know it was loaded, the funeral is not less sad or solemn.

Because the reckless young fool didn't know that the boat could be upset, the resuscitation of the victims of his folly is not less hopeless.

If a man unwittingly violates a law his plea that he *did not know* that said law existed is not sufficient defense. He must pay the same penalty as the man who deliberately breaks the statute.

Now, what in all Gaul has this to do with the price of kilowatts, as one might say?

Just this: Both man and nature agree that ignorance is no defense.

The engineer who unknowingly operates a weak boiler is just as much in danger of suddenly losing his life as is he who, knowing that a boiler is in bad shape, deliberately continues to work it.

The man who ignorantly operates an engine, pump or other machine in such a condition that it is liable suddenly to fail and thereby cause serious loss in money, time and patience, runs just as much risk of losing his job as does the man who is wilfully negligent.

The engineer who *does not know* of the leaks existing in his plant is no more excusable than he who does know, but neglects to stop them. Both are in the shadow of the "big hook."

But, why say more?

Illustration after illustration could be submitted tending to show that the consequences of lack of knowledge are far more severe than the labor involved in acquiring a sufficient amount of it.

When we speak of knowledge, we do not mean book knowledge alone, we mean such knowledge as will make you the best man for your position. This must include, then, a thorough knowledge of your plant—its capabilities and its weaknesses.

Learn the game and all its rules first. then get the philosophy.

Remember, ignorance is no defense.

# The Steam Turbine in Germany

By F. E. Junge
and E. Heinrich

*A description of the Bergmann turbine, the important features of which are: a small number of stages; a high degree of expansion before the steam enters the turbine proper; solid attachment of blades, and avoidance of the critical speed at which vibrations of the shaft are set up.*

The Bergmann Elektricitäts Gesellschaft, of Berlin, builds steam turbines of the pure impulse type, combining velocity and pressure stages. The steam enters the inlet chest at full pressure and expands in the nozzles down to a pressure of about one atmosphere. Such a high degree of expansion, permitting the formation of a compact steam jet of equivalent velocity without loss, requires the employment of conically divergent nozzles. In cylindrical nozzles the steam does not expand further than the critical ratio, which is a ratio of the pressure in front of the nozzles to that behind the nozzles and this, with dry saturated steam, is about 0.58. Whether the steam pressure before the nozzles be increased or that behind the nozzles diminished, the exit pressure will never be less than 0.58 of the initial pressure; hence the velocity of the steam will never be higher than that proportional to the critical pressure drop.

This critical velocity is practically identical with the velocity of travel of sound in steam of corresponding density; namely, about 1476 feet per second. If the pressure in the space behind the mouth of the nozzles is kept below the critical pressure, the steam emerging from the nozzle assumes the pressure of the surrounding medium; but the energy of this further expansion is entirely absorbed by the breaking up of the jet and by the formation of eddies and stationary fluctuations. A compact jet of definite direction and higher velocity can only be attained by means of a conical prolongation of the cylindrical nozzles in which the expanding steam converts the whole of its energy contents into velocity without considerable loss. By thus expanding from the boiler pressure down to the condenser pressure velocities of 4000 feet per second and higher are attained.

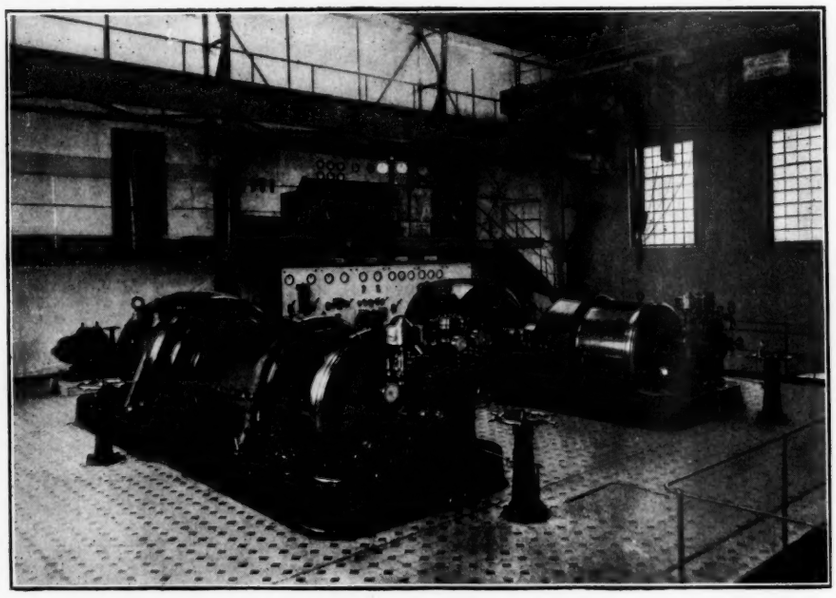
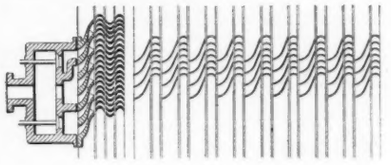


FIG. 41. TWO 1500-KILOWATT BERGMANN TURBINES

In the admission nozzles of the Bergmann turbine, expansion is carried down to one atmosphere, giving velocities of from 2600 to 3000 feet per second, and temperatures of about 300 to 340 degrees Fahrenheit. The diagrams in Fig. 42 show the relation of pressure and velocity. One row of blades being insufficient to utilize the whole velocity of the steam at normal blade speeds, the steam after leaving the first row of running blades is reversed in the following series of stationary blades and impinges upon a second row of runners at a suitable angle. In this second row the remaining velocity of the steam is utilized or absorbed, leaving just enough to effect its onward movement and issue. The process of energy conversion and also the process of regulation are the same, essentially, as in the turbines of the Allgemeine Elektricitäts Gesellschaft, previously described.

Before entering the nozzle chamber the steam passes a valve, controlled by the governor, which throttles the steam according to the requirements of the load.



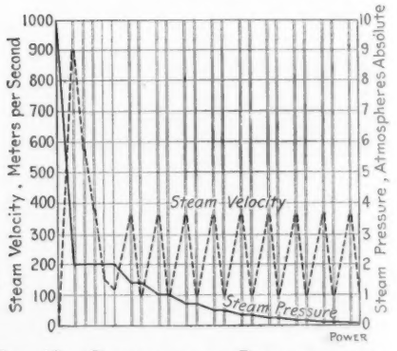



FIG. 42. RELATIONS OF PRESSURE AND VELOCITY

But as throttling involves a loss it is desirable to have the full steam pressure at all loads in front of the nozzles; therefore, when entering the nozzle chamber, the steam is made to pass a number of valves which give admission to the various groups of nozzles, each group containing a different number. By combining various groups any number of nozzles are made to operate on the turbine. In this way both the cross-section and the quantity of steam admitted are adjusted to every condition of the load and the unavoidable losses at partial loads are reduced to a minimum (see Fig. 43).

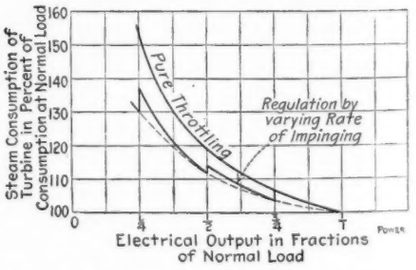

Steam Consumption of Turbine in Percent of Consumption at Normal Load
Pure Throttling
Regulation by varying Rate of Impinging
Electrical Output in Fractions of Normal Load
POWER

FIG. 43. RESULTS WITH DIFFERENT MEANS OF REGULATION

In some turbines the nozzles are distributed symmetrically over the whole circumference of the casing. But there is the disadvantage in such an arrange-

ment that the blade channels must be filled and emptied behind each nozzle, involving losses through shock and eddies, and by a symmetrical arrangement these losses are multiplied; whereas by concentrating all the nozzles into one closed segment the losses are minimized. the whole circumference of the disk. Fig. 45 shows the first running wheel with two rows of blades. The remaining pressure drop is then divided into so many stages that the velocity of the steam in each stage remains below the velocity of sound, which is the upper

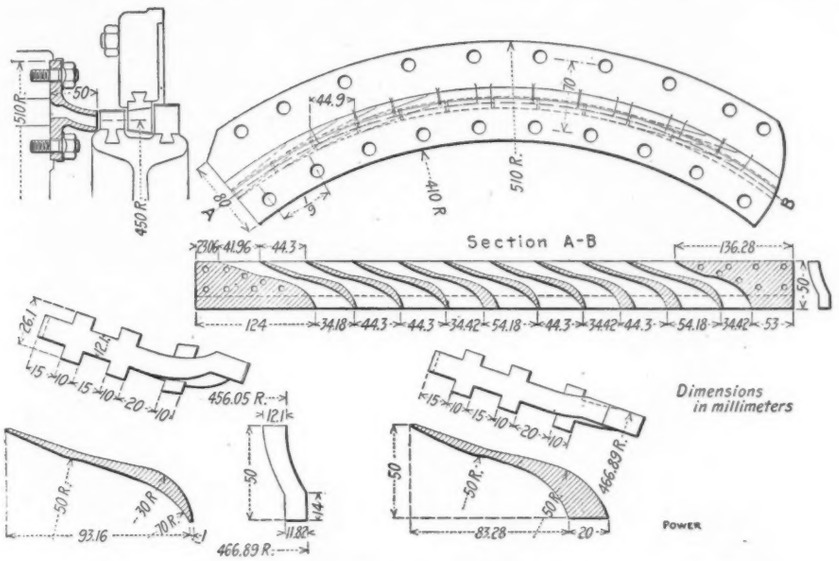



FIG. 44. DETAILS OF NOZZLE SEGMENT

Fig. 44 shows the details of a nozzle segment for a turbine of 550 kilowatts running at 3000 revolutions per minute. The latter arrangement, moreover, has the advantage that the high admission pressure and the high temperature of superheat occur only in a comparatively small section, which can be designed with

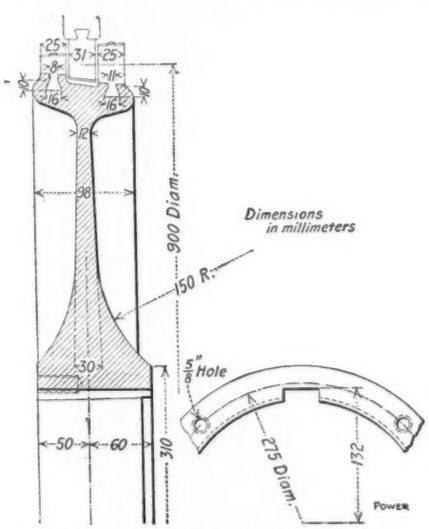



FIG. 45. FIRST RUNNING WHEEL WITH DOUBLE ROW OF BLADES

due consideration to the high pressures and temperatures without affecting the other castings of the turbine.

After leaving the second row of blades of the velocity wheel the volume of the expanded steam is so large that the succeeding blades can be distributed around

limit of the velocity attained by steam in prismatic nozzles. The latter are, of course, simpler in construction than conically divergent nozzles, and the shock and friction losses are much smaller at lower speeds. In the pressure stages, speeds range from 1000 to 1150 feet per second and can be effective-

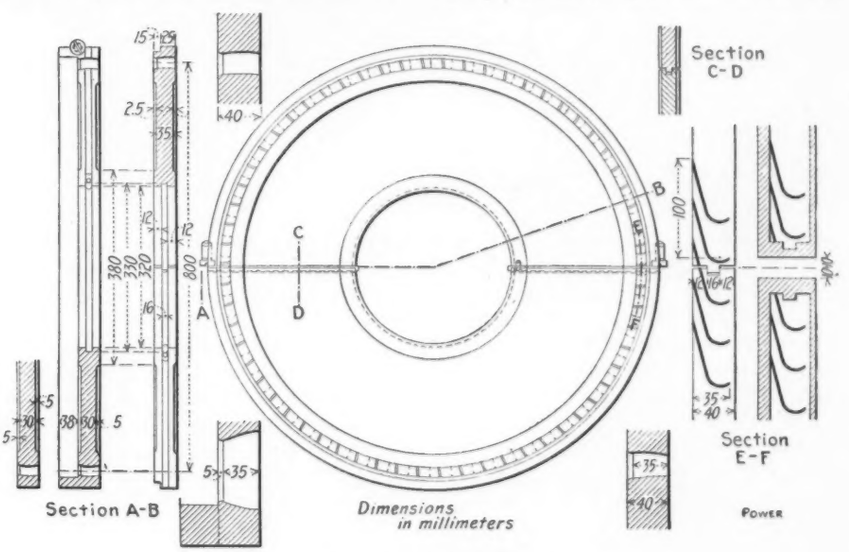



FIG. 46. GUIDE DISK

ly utilized in wheels having only one row of blades.

The guide-blade channels which serve to impart the required velocity and direction to the steam in each stage are made of nickel steel and are cast in the circumference of the guide disks (see Figs.

46 and 47). The latter are divided in the horizontal center plane so that the upper half can be lifted together with the upper part of the casing. The hubs of the running wheels are fastened to the turbine shaft in the ordinary manner and rotate within the hub boxes of the guide disks with as little play as possible and without friction. They are forged in one piece of steel with the hubs, the blades being inserted on the

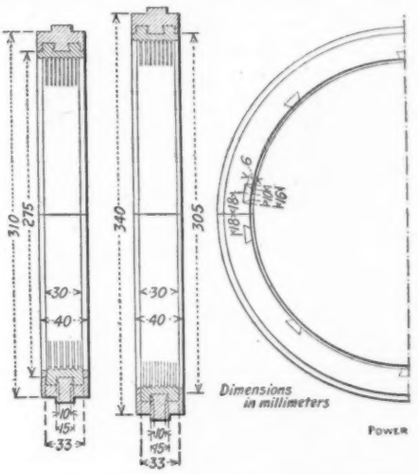



FIG. 47. RING ON STATIONARY DISK

circumference. The cross-sections are so dimensioned that the drop of pressure occurs exclusively in the guiding channels.

The pressure on both sides of the running wheels is the same so that the disks may be bored for balancing or for equalizing minor pressure differences. There is the other advantage, previously referred to, that with the same pressure before and behind the running wheels there is no axial thrust in the system, the small thrust bearing at the end of the shaft serving no other purpose than that of keeping the running wheels in their

exact position. A running wheel of the pressure stage with blades attached is shown in Fig. 48.

In order to avoid atomizing and eddying in the jet, the blade channels of the velocity wheel are accurately proportioned to the weight of steam flowing; this requires blades considerably thicker at the middle than at the edges. Until lately these blades were made of a special bronze but are now made of 25 per cent. nickel steel. In the simple pressure wheels the danger of atomizing is less imminent, wherefore the blades of nickel-steel plate are found to give satisfaction. Their method of construction is primarily dictated by the demand for light weight; yet they must be rigidly fastened on account of the stresses due to centrifugal force and possibly to

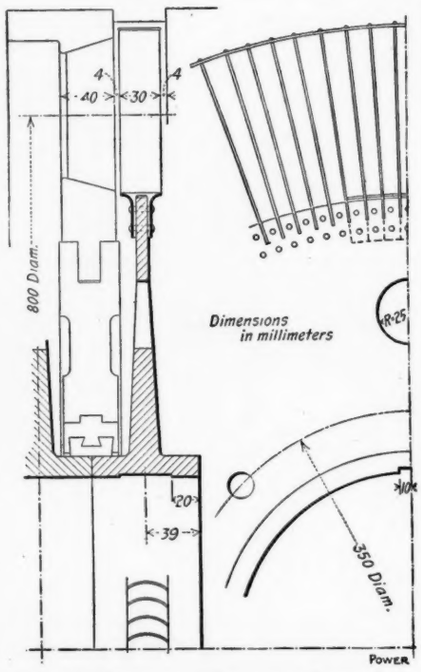



FIG. 48. RUNNING WHEEL AT PRESSURE STAGE

friction with the stationary part of the system. The attachment of the blades is a special feature of the Bergmann turbine, being covered by a German patent. Blades subjected to stress on the testing machine show a resistance to dislodgment of 4600 pounds for each blade.

The arrangement of the frame and bearings is similar to that of the Allgemeine Elektricitäts Gesellschaft turbines. The two back bearings are cast in one piece with the casing, there being no possibility of any except rotary motion between the fixed and movable parts. The front bearing is centered into and bolted on the cover of the turbine casing, which is a steel casting; hence, there is no possibility of unequal expansion through influx of heat and the unavoidable play between the fixed and rotary parts can be accurately provided for. This is of importance especially where the hubs of the running wheels pass through the bushings of the guide disks, and where the turbine shaft passes through the stuffing boxes in the heads of the casing. The radial clearance between the fixed and movable wheels is only a few thousandths of an inch; nevertheless, it is a source of loss, because the steam which passes from one side of the disk to the other renders no useful work. It is therefore important to keep this clearance not only as small as possible but also as constant as possible. The same holds true of the packing boxes.

Another reason for maintaining rigidity of construction and true concentricity of position of the fixed and rotary parts lies in the movement of the shaft. Shafts of normal dimensions making 3000 revolutions per minute and over, usually run above the critical speed. The latter corresponds to the number of revolutions at which the deflection due to the centrifugal force acting on the unbalanced masses of the shaft produces the maximum vibration. Heavy shafts running above the critical speed are apt, when passing through that speed, to vibrate badly, involving serious wear upon the bushings of the guide disks. If, in addition, there is a fault in erection or a shifting between the shaft and the casing owing to unequal expansion, the clearance may become so large that the steam consumption is increased excessively. It is desirable therefore to let the turbine run below the critical speed, even when the normal speed is 3000 revolutions per minute. This means that a shaft should be designed for a critical speed of about 4000 if one takes into consideration momentary increases and unavoidable deviations of practice from calculation.

The critical speed depends solely upon the deflection of the shaft through its own weight, plus the weight of the wheels, in such a manner that with a decreasing deflection the critical speed increases. Therefore, in order to get practical speeds the deflection of the shaft must be kept as small as possible. But the deflection grows in direct proportion to the load and as the cube of the distance between bearings; hence, the weight of the rotary part, and especially the distance between the bearings, must be kept as small as possible. The Bergmann construction of blades satisfies the first requirement of light weight, while the combination of one velocity wheel with from three to five pressure wheels results in shortening the distance between the bearings, so that the shafts are moderately heavy and the critical speed lies far above the normal. Unless these precautions are taken the actual steam consumption of turbines will be considerably higher than the consumption ascertained in shop tests, upon which guarantee figures are generally based.

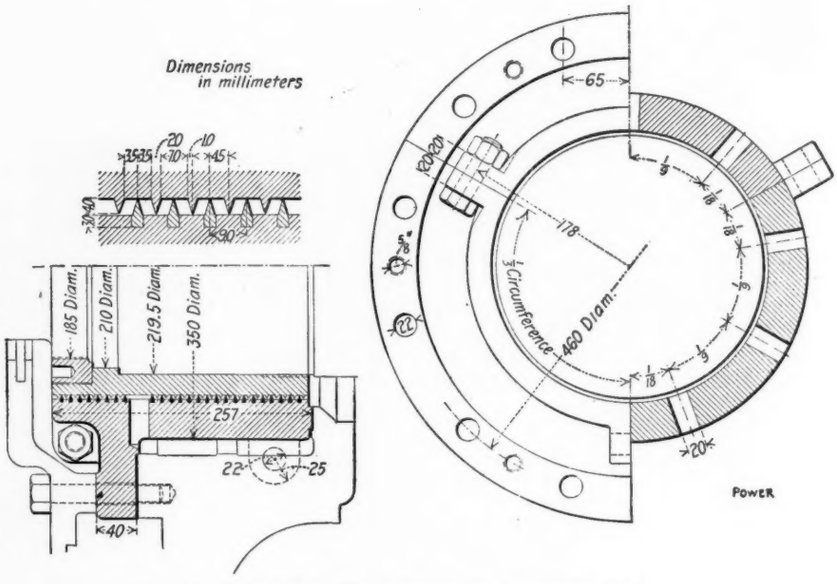



FIG. 49. DETAILS OF HIGH-PRESSURE STUFFING BOX

The packing of the Bergmann turbine is of the labyrinth variety, both on the high-pressure side, where the pressure is about 15 pounds above the atmosphere, and on the low-pressure side, where the packing separates the vacuum space from the atmosphere. The rings on the cover which project into the annular grooves of the bushings on the shaft are divided into two groups of different sizes, the space between them connecting from the high-pressure to the low-pressure stuffing boxes. Details of the high-pressure stuffing box are shown in Fig. 49. Thus the steam emerging from the first is used in the second as a packing medium against the influx of air from without. At light loads, when there is not sufficient surplus steam in the high-pressure box, live steam can be introduced into the connecting pipe, while at heavy loads the surplus steam not

utilized in the low-pressure packing box is discharged into one of the middle stages of the turbine.

The construction of the bearings does not differ from the ordinary. They are lubricated with oil which is supplied under about two atmospheres pressure by a rotary pump. After passing the bearings the oil is collected in a tank where it is filtered and used over again. The bearings are cooled partly by means of a water-jacketed worm pipe inserted between the oil pump and the bearings, and partly by water cooling the bearings themselves.

The same worm gear which drives the oil pump also serves to actuate the vertical shaft of the governor. This is of the Hartung type and acts indirectly on the main admission valve through a balanced pilot valve. The latter admits oil under pressure below or above the main governing piston which is fastened on the same stem with the throttle valve. In addition to this automatic regulation there is, within certain load ranges, regulation by hand, inasmuch as one or more groups of nozzles may be closed so as to obtain the benefit of the full steam pressure behind the throttle valve and therefore in front of the inlet nozzles. In addition there is the customary safety regulator closing the main admission valve automatically when the speed exceeds certain limits.

Summarizing the notable features of the Bergmann turbines, they will be found fairly representative of the standard construction of steam turbines in Germany; and are as follows: Far-reaching expansion and cooling of the steam before it enters the turbine proper, whereby moderate temperatures in the casing are secured; the smallest number of stages compatible with moderate speed, therefore short length of machine; avoidance of the critical speed and of vibration and other troubles connected therewith; the construction of the casing and bearings in one piece on a common frame plate and accurate concentric position of the fixed and movable parts, whereby good steam economy is guaranteed; no lubrication of packings and therefore no oil in the condensate; solid attachment of blades, which can be exchanged without having to dismount the turbine; and great resistance of blades against bending and shearing. Fig. 50 shows a longitudinal section through the complete turbine.

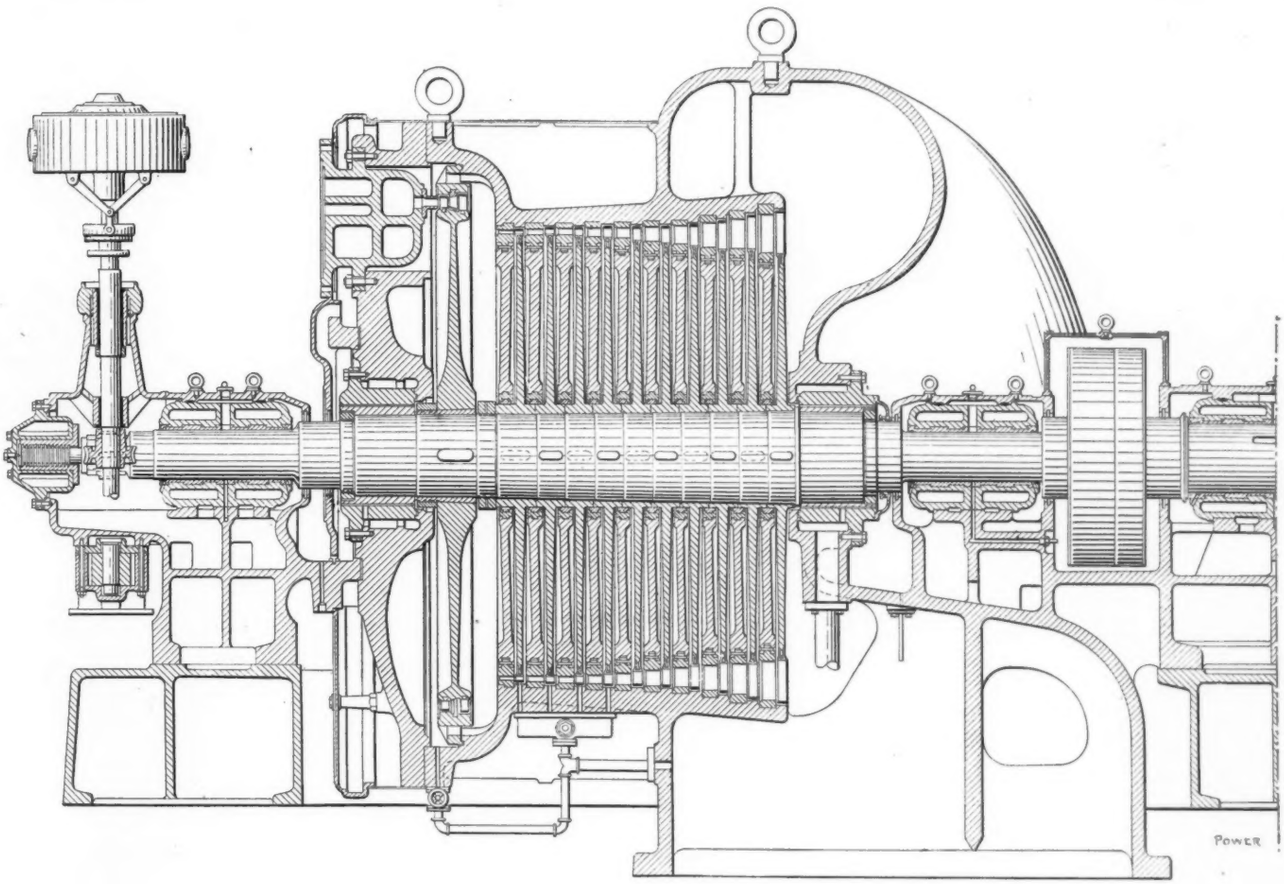



FIG. 50. SECTIONAL ELEVATION OF BERGMANN TURBINE

A mixture of Pintsch gas with air is explosive within a gas percentage range of 5 to 13 per cent., and it develops its maximum power at a percentage of about 9 per cent.

## Metal Welding in Germany

Consul-General A. M. Thackara, of Berlin, states that there are many systems of metal-welding apparatus made and used in Germany. The use of welding apparatus has very materially increased in recent times. The quite general replacing of the older hydrogen by the acetylene apparatus makes the process not only cheaper, but also more generally applicable. The calcium carbide is readily obtainable, and its generation is very simple.

In the case of the latter system, the acetylene gas is generated directly from calcium carbide by the apparatus itself. The cost of acetylene gas thus produced is about the same as the market price for hydrogen gas, but only about one-fifth as much acetylene gas is required for a given piece of welding. Furthermore, the considerably higher temperature attained with acetylene gas makes possible the welding of metals of greater thickness. The temperature limit for hydrogen is 1900 degrees Centigrade (3452 degrees Fahrenheit) and of acetylene it is 3500 degrees Centigrade (6332 degrees Fahrenheit). The metal thickness that may be welded by the two systems are ⅓ and 1⅛ inches, respectively, leaving a margin of ⅞ inch in favor of the acetylene gas.

# Schedule of Flanged Fittings

The committee on standardization of the society working in conjunction with a similar committee of the National Association of Master Steam and Hot Water Fitters presented the following report on standard weight and extra-heavy flanged fittings at the spring meeting of the American Society of Mechanical Engineers:

Standard or extra-heavy reducing elbows carry the same dimensions center to face as the regular elbows of the largest straight size.

Standard or extra-heavy tees, crosses, laterals or Y-branches, reducing on run or outlets, carry the same dimensions face to face and center to face as the largest straight size.

If flanged fittings for lower working pressures than 125 pounds are made they shall conform in all dimensions except thickness of shell to this standard, and guaranteed working pressure must be cast on each fitting.

Companion flanges for these fittings must be standard dimensions.

Where long-turn fittings are specified

TABLE 1. SCHEDULE OF STANDARD WEIGHT FLANGED FITTINGS

| | Elbows | 45-Deg. Elbows | Long Turn Elbows | Tees and Cros'es | Laterals or Y Br'n'h's |
|---|---|---|---|---|---|
| **1¼ inch** | | | | | |
| Center to face | 3¼ | 2 | 6 | 3¼ | 6½ |
| Face to face | .... | .... | .... | 7½ | 8½ |
| Diameter of flange | 4½ | 4½ | 4½ | 4½ | 4½ |
| Thickness of flange | 7/16 | 7/16 | 7/16 | 7/16 | 7/16 |
| Diameter of bolt circle | 3⅜ | 3⅜ | 3⅜ | 3⅜ | 3⅜ |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | 9/16 | 9/16 | 9/16 | 9/16 | 9/16 |
| **1½ inch** | | | | | |
| Center to face | 4 | 2¼ | 6½ | 4 | 6⅞ |
| Face to face | .... | .... | .... | 8 | 9¼ |
| Diameter of flange | 5 | 5 | 5 | 5 | 5 |
| Thickness of flange | 9/16 | 9/16 | 9/16 | 9/16 | 9/16 |
| Diameter of bolt circle | 3⅞ | 3⅞ | 3⅞ | 3⅞ | 3⅞ |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | ⅝ | ⅝ | ⅝ | ⅝ | ⅝ |
| **2 inch** | | | | | |
| Center to face | 4½ | 2½ | 7 | 4½ | 8 |
| Face to face | .... | .... | .... | 9 | 10½ |
| Diameter of flange | 6 | 6 | 6 | 6 | 6 |
| Thickness of flange | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 |
| Diameter of bolt circle | 4¾ | 4¾ | 4¾ | 4¾ | 4¾ |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | ¾ | ¾ | ¾ | ¾ | ¾ |
| **2½ inch** | | | | | |
| Center to face | 5 | 2¾ | 7½ | 5 | 9½ |
| Face to face | .... | .... | .... | 10 | 12 |
| Diameter of flange | 7 | 7 | 7 | 7 | 7 |
| Thickness of flange | 11/16 | 11/16 | 11/16 | 11/16 | 11/16 |
| Diameter of bolt circle | 5½ | 5½ | 5½ | 5½ | 5½ |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | ¾ | ¾ | ¾ | ¾ | ¾ |
| **3 inch** | | | | | |
| Center to face | 5½ | 3 | 8 | 5½ | 10 |
| Face to face | .... | .... | .... | 11 | 13 |
| Diameter of flange | 7½ | 7½ | 7½ | 7½ | 7½ |
| Thickness of flange | 13/16 | 13/16 | 13/16 | 13/16 | 13/16 |
| Diameter of bolt circle | 6 | 6 | 6 | 6 | 6 |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | ¾ | ¾ | ¾ | ¾ | ¾ |
| **3½ inch** | | | | | |
| Center to face | 6 | 3½ | 9 | 6 | 11½ |
| Face to face | .... | .... | .... | 12 | 14½ |
| Diameter of flange | 8½ | 8½ | 8½ | 8½ | 8½ |
| Thickness of flange | 13/16 | 13/16 | 13/16 | 13/16 | 13/16 |
| Diameter of bolt circle | 7 | 7 | 7 | 7 | 7 |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | ¾ | ¾ | ¾ | ¾ | ¾ |
| **4 inch** | | | | | |
| Center to face | 6½ | 4 | 10 | 6½ | 12 |
| Face to face | .... | .... | .... | 13 | 15 |
| Diameter of flange | 9 | 9 | 9 | 9 | 9 |
| Thickness of flange | 15/16 | 15/16 | 15/16 | 15/16 | 15/16 |
| Diameter of bolt circle | 7½ | 7½ | 7½ | 7½ | 7½ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |
| **4½ inch** | | | | | |
| Center to face | 7 | 4½ | 11 | 7 | 12½ |
| Face to face | .... | .... | .... | 14 | 15½ |
| Diameter of flange | 9½ | 9½ | 9½ | 9½ | 9½ |
| Thickness of flange | 15/16 | 15/16 | 15/16 | 15/16 | 15/16 |
| Diameter of bolt circle | 7¼ | 7¼ | 7¼ | 7¼ | 7¼ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |
| **5 inch** | | | | | |
| Center to face | 7½ | 4½ | 12 | 7½ | 13½ |
| Face to face | .... | .... | .... | 15 | 17 |
| Diameter of flange | 10 | 10 | 10 | 10 | 10 |
| Thickness of flange | 15/16 | 15/16 | 15/16 | 15/16 | 15/16 |
| Diameter of bolt circle | 8½ | 8½ | 8½ | 8½ | 8½ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |
| **6 inch** | | | | | |
| Center to face | 8 | 5 | 13 | 8 | 14½ |
| Face to face | .... | .... | .... | 16 | 18 |
| Diameter of flange | 11 | 11 | 11 | 11 | 11 |
| Thickness of flange | 1 | 1 | 1 | 1 | 1 |
| Diameter of bolt circle | 9½ | 9½ | 9½ | 9½ | 9½ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |
| **7 inch** | | | | | |
| Center to face | 8½ | 5½ | 14½ | 8½ | 16½ |
| Face to face | .... | .... | .... | 17 | 20½ |
| Diameter of flange | 12½ | 12½ | 12½ | 12½ | 12½ |
| Thickness of flange | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 |
| Diameter of bolt circle | 10¼ | 10¼ | 10¼ | 10¼ | 10¼ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |

| | Elbows | 45-Deg. Elbows | Long Turn Elbows | Tees and Cros'es | Laterals or Y Br'n'h's |
|---|---|---|---|---|---|
| **8 inch** | | | | | |
| Center to face | 9 | 6 | 16 | 9 | 17½ |
| Face to face | .... | .... | .... | 18 | 22 |
| Diameter of flange | 13½ | 13½ | 13½ | 13½ | 13½ |
| Thickness of flange | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| Diameter of bolt circle | 11¼ | 11¼ | 11¼ | 11¼ | 11¼ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |
| **9 inch** | | | | | |
| Center to face | 10 | 6½ | 18 | 10 | 19½ |
| Face to face | .... | .... | .... | 20 | 24 |
| Diameter of flange | 15 | 15 | 15 | 15 | 15 |
| Thickness of flange | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| Diameter of bolt circle | 13¼ | 13¼ | 13¼ | 13¼ | 13¼ |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | ⅞ | ⅞ | ⅞ | ⅞ | ⅞ |
| **10 inch** | | | | | |
| Center to face | 11 | 7 | 20 | 11 | 20½ |
| Face to face | .... | .... | .... | 22 | 25½ |
| Diameter of flange | 16 | 16 | 16 | 16 | 16 |
| Thickness of flange | 1 3/16 | 1 3/16 | 1 3/16 | 1 3/16 | 1 3/16 |
| Diameter of bolt circle | 14¼ | 14¼ | 14¼ | 14¼ | 14¼ |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | 1 | 1 | 1 | 1 | 1 |
| **12 inch** | | | | | |
| Center to face | 12 | 7½ | 22 | 12 | 24½ |
| Face to face | .... | .... | .... | 24 | 30 |
| Diameter of flange | 19 | 19 | 19 | 19 | 19 |
| Thickness of flange | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |
| Diameter of bolt circle | 17 | 17 | 17 | 17 | 17 |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | 1 | 1 | 1 | 1 | 1 |
| **14 inch** | | | | | |
| Center to face | 14 | 7½ | 24 | 14 | 27 |
| Face to face | .... | .... | .... | 28 | 33 |
| Diameter of flange | 21 | 21 | 21 | 21 | 21 |
| Thickness of flange | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| Diameter of bolt circle | 18¾ | 18¾ | 18¾ | 18¾ | 18¾ |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| **15 inch** | | | | | |
| Center to face | 14½ | 8 | 26 | 14½ | 28½ |
| Face to face | .... | .... | .... | 29 | 34½ |
| Diameter of flange | 22½ | 22½ | 22½ | 22½ | 22½ |
| Thickness of flange | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| Diameter of bolt circle | 20 | 20 | 20 | 20 | 20 |
| Number of bolt holes | 16 | 16 | 16 | 16 | 16 |
| Diameter of bolt holes | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| **16 inch** | | | | | |
| Center to face | 15 | 8 | 28 | 15 | 30 |
| Face to face | .... | .... | .... | 30 | 36½ |
| Diameter of flange | 23½ | 23½ | 23½ | 23½ | 23½ |
| Thickness of flange | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 |
| Diameter of bolt circle | 21¼ | 21¼ | 21¼ | 21¼ | 21¼ |
| Number of bolt holes | 16 | 16 | 16 | 16 | 16 |
| Diameter of bolt holes | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| **18 inch** | | | | | |
| Center to face | 16½ | 8½ | 30 | 16½ | 32 |
| Face to face | .... | .... | .... | 33 | 39 |
| Diameter of flange | 25 | 25 | 25 | 25 | 25 |
| Thickness of flange | 1 3/16 | 1 3/16 | 1 3/16 | 1 3/16 | 1 3/16 |
| Diameter of bolt circle | 22¾ | 22¾ | 22¾ | 22¾ | 22¾ |
| Number of bolt holes | 16 | 16 | 16 | 16 | 16 |
| Diameter of bolt holes | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |
| **20 inch** | | | | | |
| Center to face | 18 | 9½ | 32 | 18 | 35 |
| Face to face | .... | .... | .... | 36 | 43 |
| Diameter of flange | 27½ | 27½ | 27½ | 27½ | 27½ |
| Thickness of flange | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 |
| Diameter of bolt circle | 25 | 25 | 25 | 25 | 25 |
| Number of bolt holes | 20 | 20 | 20 | 20 | 20 |
| Diameter of bolt holes | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |
| **22 inch** | | | | | |
| Center to face | 20 | 10 | 34 | 20 | 37½ |
| Face to face | .... | .... | .... | 40 | 46 |
| Diameter of flange | 29½ | 29½ | 29½ | 29½ | 29½ |
| Thickness of flange | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 |
| Diameter of bolt circle | 27¼ | 27¼ | 27¼ | 27¼ | 27¼ |
| Number of bolt holes | 20 | 20 | 20 | 20 | 20 |
| Diameter of bolt holes | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |
| **24 inch** | | | | | |
| Center to face | 22 | 11 | 36 | 22 | 40½ |
| Face to face | .... | .... | .... | 44 | 49½ |
| Diameter of flange | 32 | 32 | 32 | 32 | 32 |
| Thickness of flange | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| Diameter of bolt circle | 29½ | 29½ | 29½ | 29½ | 29½ |
| Number of bolt holes | 20 | 20 | 20 | 20 | 20 |
| Diameter of bolt holes | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |

it has reference only to elbows, which are made in two dimensions, to be known as "elbows" and "long-turn elbows," the latter being used only when so specified.

All standard-weight fittings must be guaranteed for 125 pounds and extra-heavy fittings for 250 pounds working pressure, and each fitting must have some mark cast on it indicating the maker and guaranteed working steam pressure.

All extra-heavy fittings and flanges to have a raised surface 1/16 inch high inside of the bolt holes for a gasket.

Standard-weight fittings and flanges to be plain faced.

Bolts to be 1/8 inch smaller in diameter than bolt holes.

Size of all fittings scheduled indicates inside diameter of ports.

For outside dimension pipe use corresponding size of inside dimension fittings.

These dimensions, both standard and extra heavy, refer to either cast-iron or cast-steel fittings and rolled-steel flanges.

The committee consists of H. G. Stott, chairman, superintendent of motive power of the Interborough; F. R. Hutton, past president of the Society; I. E. Moulthrop, of the Boston Edison; H. P. Norton, Bureau of Steam Engineering, Navy Department, and J. T. Whittlesey, of the Public Service Corporation of New Jersey.

TABLE 2. SCHEDULE OF EXTRA HEAVY FLANGED FITTINGS

| | Elbows | 45-Deg. Elbows | Long Turn Elbows | Tees and Cros'es | Laterals or Y Br'n'h's |
|---|---|---|---|---|---|
| 1¼ inch | | | | | |
| Center to face | 4½ | 2½ | 6 | 4½ | 7½ |
| Face to face | .... | .... | .... | 8½ | 9¾ |
| Diameter of flange | 5 | 5 | 5 | 5 | 5 |
| Thickness of flange | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 |
| Diameter of bolt circle | 3¾ | 3¾ | 3¾ | 3¾ | 3¾ |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | 5/8 | 5/8 | 5/8 | 5/8 | 5/8 |
| 1½ inch | | | | | |
| Center to face | 4½ | 2¾ | 6½ | 4½ | 8½ |
| Face to face | .... | .... | .... | 9 | 11 |
| Diameter of flange | 6 | 6 | 6 | 6 | 6 |
| Thickness of flange | 13/16 | 13/16 | 13/16 | 13/16 | 13/16 |
| Diameter of bolt circle | 4½ | 4½ | 4½ | 4½ | 4½ |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 |
| 2 inch | | | | | |
| Center to face | 5 | 3 | 7 | 5 | 9 |
| Face to face | .... | .... | .... | 10 | 11½ |
| Diameter of flange | 6½ | 6½ | 6½ | 6½ | 6½ |
| Thickness of flange | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| Diameter of bolt circle | 5 | 5 | 5 | 5 | 5 |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | 3/4 | 3/4 | 3/4 | 3/4 | 3/4 |
| 2½ inch | | | | | |
| Center to face | 5½ | 3½ | 7½ | 5½ | 10½ |
| Face to face | .... | .... | .... | 11 | 13 |
| Diameter of flange | 7½ | 7½ | 7½ | 7½ | 7½ |
| Thickness of flange | 1 | 1 | 1 | 1 | 1 |
| Diameter of bolt circle | 5⅞ | 5⅞ | 5⅞ | 5⅞ | 5⅞ |
| Number of bolt holes | 4 | 4 | 4 | 4 | 4 |
| Diameter of bolt holes | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| 3 inch | | | | | |
| Center to face | 6 | 3½ | 8 | 6 | 11 |
| Face to face | .... | .... | .... | 12 | 14 |
| Diameter of flange | 8¼ | 8¼ | 8¼ | 8¼ | 8¼ |
| Thickness of flange | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| Diameter of bolt circle | 6⅝ | 6⅝ | 6⅝ | 6⅝ | 6⅝ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| 3½ inch | | | | | |
| Center to face | 6½ | 4 | 9 | 6½ | 12½ |
| Face to face | .... | .... | .... | 13 | 15½ |
| Diameter of flange | 9 | 9 | 9 | 9 | 9 |
| Thickness of flange | 1 3/16 | 1 3/16 | 1 3/16 | 1 3/16 | 1 3/16 |
| Diameter of bolt circle | 7¼ | 7¼ | 7¼ | 7¼ | 7¼ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| 4 inch | | | | | |
| Center to face | 7 | 4½ | 10 | 7 | 13½ |
| Face to face | .... | .... | .... | 14 | 16½ |
| Diameter of flange | 10 | 10 | 10 | 10 | 10 |
| Thickness of flange | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |
| Diameter of bolt circle | 7⅞ | 7⅞ | 7⅞ | 7⅞ | 7⅞ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| 4½ inch | | | | | |
| Center to face | 7½ | 4½ | 11 | 7½ | 14½ |
| Face to face | .... | .... | .... | 15 | 18 |
| Diameter of flange | 10½ | 10½ | 10½ | 10½ | 10½ |
| Thickness of flange | 1 5/16 | 1 5/16 | 1 5/16 | 1 5/16 | 1 5/16 |
| Diameter of bolt circle | 8½ | 8½ | 8½ | 8½ | 8½ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| 5 inch | | | | | |
| Center to face | 8 | 5 | 12 | 8 | 15 |
| Face to face | .... | .... | .... | 16 | 18½ |
| Diameter of flange | 11 | 11 | 11 | 11 | 11 |
| Thickness of flange | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| Diameter of bolt circle | 9¼ | 9¼ | 9¼ | 9¼ | 9¼ |
| Number of bolt holes | 8 | 8 | 8 | 8 | 8 |
| Diameter of bolt holes | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| 6 inch | | | | | |
| Center to face | 9 | 5½ | 13 | 9 | 17½ |
| Face to face | .... | .... | .... | 18 | 21½ |
| Diameter of flange | 12½ | 12½ | 12½ | 12½ | 12½ |
| Thickness of flange | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 | 1 1/16 |
| Diameter of bolt circle | 10⅝ | 10⅝ | 10⅝ | 10⅝ | 10⅝ |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | 7/8 | 7/8 | 7/8 | 7/8 | 7/8 |
| 7 inch | | | | | |
| Center to face | 9½ | 5¾ | 14½ | 9½ | 18½ |
| Face to face | .... | .... | .... | 19 | 23 |
| Diameter of flange | 14 | 14 | 14 | 14 | 14 |
| Thickness of flange | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| Diameter of bolt circle | 11⅞ | 11⅞ | 11⅞ | 11⅞ | 11⅞ |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | 1 | 1 | 1 | 1 | 1 |

| | Elbows | 45-Deg. Elbows | Long Turn Elbows | Tees and Cros'es | Laterals or Y Br'n'h's |
|---|---|---|---|---|---|
| 8 inch | | | | | |
| Center to face | 10½ | 6 | 16 | 10½ | 20 |
| Face to face | .... | .... | .... | 21 | 25 |
| Diameter of flange | 15 | 15 | 15 | 15 | 15 |
| Thickness of flange | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| Diameter of bolt circle | 13 | 13 | 13 | 13 | 13 |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | 1 | 1 | 1 | 1 | 1 |
| 9 inch | | | | | |
| Center to face | 11 | 6½ | 18 | 11 | 21 |
| Face to face | .... | .... | .... | 22 | 26 |
| Diameter of flange | 16 | 16 | 16 | 16 | 16 |
| Thickness of flange | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| Diameter of bolt circle | 14 | 14 | 14 | 14 | 14 |
| Number of bolt holes | 12 | 12 | 12 | 12 | 12 |
| Diameter of bolt holes | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| 10 inch | | | | | |
| Center to face | 12 | 7 | 20 | 12 | 23 |
| Face to face | .... | .... | .... | 24 | 28½ |
| Diameter of flange | 17½ | 17½ | 17½ | 17½ | 17½ |
| Thickness of flange | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| Diameter of bolt circle | 15¼ | 15¼ | 15¼ | 15¼ | 15¼ |
| Number of bolt holes | 16 | 16 | 16 | 16 | 16 |
| Diameter of bolt holes | 1⅛ | 1⅛ | 1⅛ | 1⅛ | 1⅛ |
| 12 inch | | | | | |
| Center to face | 13½ | 8 | 22 | 13½ | 26½ |
| Face to face | .... | .... | .... | 27 | 32½ |
| Diameter of flange | 20 | 20 | 20 | 20 | 20 |
| Thickness of flange | 2 | 2 | 2 | 2 | 2 |
| Diameter of bolt circle | 17¾ | 17¾ | 17¾ | 17¾ | 17¾ |
| Number of bolt holes | 16 | 16 | 16 | 16 | 16 |
| Diameter of bolt holes | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |
| 14 inch | | | | | |
| Center to face | 14½ | 8 | 24 | 14½ | 30 |
| Face to face | .... | .... | .... | 29 | 36½ |
| Diameter of flange | 23 | 23 | 23 | 23 | 23 |
| Thickness of flange | 2⅛ | 2⅛ | 2⅛ | 2⅛ | 2⅛ |
| Diameter of bolt circle | 20 | 20 | 20 | 20 | 20 |
| Number of bolt holes | 20 | 20 | 20 | 20 | 20 |
| Diameter of bolt holes | 1¼ | 1¼ | 1¼ | 1¼ | 1¼ |
| 15 inch | | | | | |
| Center to face | 15 | 8½ | 26 | 15 | 31½ |
| Face to face | .... | .... | .... | 30 | 38 |
| Diameter of flange | 24 | 24 | 24 | 24 | 24 |
| Thickness of flange | 2 3/16 | 2 3/16 | 2 3/16 | 2 3/16 | 2 3/16 |
| Diameter of bolt circle | 21 | 21 | 21 | 21 | 212 |
| Number of bolt holes | 20 | 20 | 20 | 20 | 20 |
| Diameter of bolt holes | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| 16 inch | | | | | |
| Center to face | 16 | 9 | 28 | 16 | 33 |
| Face to face | .... | .... | .... | 32 | 40½ |
| Diameter of flange | 26 | 26 | 26 | 26 | 26 |
| Thickness of flange | 2⅛ | 2⅛ | 2⅛ | 2⅛ | 2⅛ |
| Diameter of bolt circle | 22½ | 22½ | 22½ | 22½ | 22½ |
| Number of bolt holes | 20 | 20 | 20 | 20 | 20 |
| Diameter of bolt holes | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| 18 inch | | | | | |
| Center to face | 17 | 9½ | 30 | 17 | 36 |
| Face to face | .... | .... | .... | 34 | 44 |
| Diameter of flange | 28 | 28 | 28 | 28 | 28 |
| Thickness of flange | 2⅜ | 2⅜ | 2⅜ | 2⅜ | 2⅜ |
| Diameter of bolt circle | 24½ | 24½ | 24½ | 24½ | 24½ |
| Number of bolt holes | 24 | 24 | 24 | 24 | 24 |
| Diameter of bolt holes | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| 20 inch | | | | | |
| Center to face | 18½ | 10 | 32 | 18½ | 38 |
| Face to face | .... | .... | .... | 37 | 46½ |
| Diameter of flange | 30½ | 30½ | 30½ | 30½ | 30½ |
| Thickness of flange | 2½ | 2½ | 2½ | 2½ | 2½ |
| Diameter of bolt circle | 26¾ | 26¾ | 26¾ | 26¾ | 26¾ |
| Number of bolt holes | 24 | 24 | 24 | 24 | 24 |
| Diameter of bolt holes | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| 22 inch | | | | | |
| Center to face | 20 | 10½ | 34 | 20 | 41½ |
| Face to face | .... | .... | .... | 40 | 51 |
| Diameter of flange | 32½ | 32½ | 32½ | 32½ | 32½ |
| Thickness of flange | 2⅝ | 2⅝ | 2⅝ | 2⅝ | 2⅝ |
| Diameter of bolt circle | 28¾ | 28¾ | 28¾ | 28¾ | 28¾ |
| Number of bolt holes | 28 | 28 | 28 | 28 | 28 |
| Diameter of bolt holes | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |
| 24 inch | | | | | |
| Center to face | 22 | 11½ | 36 | 22 | 44½ |
| Face to face | .... | .... | .... | 44 | 54½ |
| Diameter of flange | 35 | 35 | 35 | 35 | 35 |
| Thickness of flange | 2¾ | 2¾ | 2¾ | 2¾ | 2¾ |
| Diameter of bolt circle | 31¼ | 31¼ | 31¼ | 31¼ | 31¼ |
| Number of bolt holes | 28 | 28 | 28 | 28 | 20 |
| Diameter of bolt holes | 1⅜ | 1⅜ | 1⅜ | 1⅜ | 1⅜ |

# Training Mr. Duffy

By Danny Hogan

"Daly do be trainin' me," said Duffy, "how to set a slide valve."

"An' what," asked Doolin, "is a slide valve? Ye have a cylinder an' a piston —to let in the steam and to let it out again at the right instant is all 'tis for. 'Tis easier than layin' out an 18-inch stack. Them engineers, Duffy, get worked up over simple matters and get stuck on others. Ask Daly when you meet him how would he indicate a turbine engine an' tell me what he says."

"About the butt-strap joints," said Duffy, "Daly said the City Hall bunch would ask all kinds of questions, as this biler matter is considered greater than the engine."

"An' right he is for once," replied Doolin. "The steam biler is the most important machine in the world. The lap-joint seam is discarded in respectable society as ondecent and the butt joint is now the rage. To explain why, Duffy, 'tis only necessary to say the lap seam is not a true circle and when pressure is on it results in a bendin' action along the seam. As the pressure varies, so does the bendin'. Ye may bend a piece of steel back and forth but at last it will break. Now the butt joint, properly made, is a true circle and hence the bendin' action is absent. To avoid bendin' stresses the shell must have all parts truly cylindrical. It's a simple truth, is it not? Now to rivet the ends of the circle together we must have straps—one inside and one outside. The common rule is to have each strap 1/16 inch less in thickness than the plate, and this is safe practice. The common practice also makes the pitch in the outer row either double or four times that in the inner row or rows. Ye noted in the table I gave you that a steel rivet in single shear is allowed 42,000 pounds per square inch and in double shear 78,000 pounds, for in the latter the rivet would be cut in half in two places and but one in the lap seam. Ye would think one could use very much smaller rivets with a butt joint as the rivets are near double as strong, but, mind ye, in all the problem a tight job is needed. True, we don't need the diameter to be

$$T \times 2,$$

but within ordinary practice we find this a good rule:

$T \times 2 - 1/16$ *inch* = *diameter of rivet hole*

and gives a tight job at that.

"Suppose, now, ye have a ½-inch plate of 56,000 pounds tensile strength, an' you want a butt-strap joint triple riveted. The straps would each be 7/16 inch thick and the rivet hole 15/16 inch. In a section of the joint there would be four rivets in double shear and one in single shear an' the shear value is

$53{,}843 \times 4 + 28{,}988 = 243{,}560$ *pounds*

"Now note, Duffy, in the single-shear joint the value of the rivet is less than the tensile strength of the plate and hence must be taken in account. But in the butt joint commonly used the shear exceeds the plate value. Therefore, we need only find a section in the outer row of rivets to give us the desired efficiency. Suppose we want an 86 per cent. joint, the outer row is the weakest section at the net plate, as the shearing value is high. If

$$\frac{P - D}{P} = \textit{efficiency of plate}$$

then

$$\frac{\textit{Diameter}}{1 - \textit{efficiency}} = \textit{pitch}$$

The diameter is 0.9375 and

$$\frac{0.9375}{1 - 0.86} = 6.696 \textit{ inches}$$

Call it 6¾ inches as the nearest common fraction. Then,

$$\frac{6.75}{6.75 - 0.9375} = 86.1 \textit{ per cent.}$$

The value of the solid plate is

$6.75 \times 0.5 \times 56{,}000 = 189{,}000$ *pounds*

The shear of all the rivets is 243,560 pounds.

"This rule applies on a double, triple or quadruple joint provided, of course, ye use, within reason, the proper size rivet. The rivets in double shear will have a pitch either one-half or one-fourth of the outer row of rivets, dependin' on whether the joint is double, triple or quadruple, for by the pitch found by this rule is meant always the outer row. For instance, say ye want the ½-inch plate of 56,000 pounds tensile strength with a butt, quadruple-riveted joint with an efficiency of 94 per cent. Then,

$$\frac{0.9375}{1 - 0.94} = 15.625 \textit{ inches}$$

for the outer row. The next row is 7.8125 inches and the inner row, the single pitch, is 3.90625 inches. In a double-riveted butt joint say we want an 80 per cent. efficiency. Then,

$$\frac{0.9375}{1 - 0.80} = 4.6875 \textit{ inches}$$

the outer pitch, the inner being 2.34375 inches, or $4\frac{11}{16}$ inches and $2\frac{11}{32}$ inches.

"Another method is to find the pitch ratio an' multiply this by the diameter of the rivet hole. The formula, Duffy, is

$$\frac{1}{1 - \textit{efficiency}} = \textit{pitch ratio}$$

for any thickness of plate.

"Suppose an 80 per cent. joint is wanted. Then,

$$\frac{1}{1 - 0.80} = 5$$

and

$$0.9375 \times 5 = 4.6875 \textit{ inches}$$

the desired pitch for this efficiency and this rivet hole in the outer row, or the double pitch. Mind you, Duffy, the double shear is greater than the tensile strength—and by finding the total shear an' comparing this with a pitch section of the solid plate you may design with any thickness of plate and any tensile strength, for any rivet the bull will drive. But, in shops not building Scotch marine bilers with heavy plate, they find it impracticable to drive, with a 100-ton bull, rivets much above 1¼ inches an' get a tight job. The marine shops require bulls up to 150 tons, an' ordinary shops stop at 100 tons. The rivet must fill the hole an' be allowed to shrink under pressure. True, a steel rivet don't swell when heated as an iron one did, an' in some shops the hole is but 1/32 inch smaller than the rivet instead of 1/16 inch. In this case it's easier to fill the hole."

"An' how about the distance between the pitch lines in a butt joint?" asked Duffy.

"Well," replied Doolin, "ye get good results by multiplying the pitch by 0.65 to get the distance between the rows in the inner rows of rivets in double shear. Mind that the plate edge should be planed square so that it butts solid together. This is important in allowing 1½ diameters from the edge of the plate to the first pitch line. For the other distances from pitch line to edge of plate, see you have the same allowance. For the rivets in single shear, lay out so the rivet head will clear the outer strap an' leave clearance for the die. Mind, in laying out the pitch by these methods 'tis better to have the diameter of rivet

$$T \times 2 - 1/16 \textit{ inch}$$

an' the straps

$$T - 1/16 \textit{ inch}$$

But, in ¼-inch plate make the straps ¼ inch, so they won't buckle and bulge when bull riveting. Indeed, it's a disputed argument among intelligent bilermakers about the thickness of the inner strap, as some claim it should be the thickness of the plate, owing to the stresses it must carry. Be that as it may, one thing I hold true, the straps should be absolutely true arcs of their respective circles in order that bending action will be eliminated. Another thing, Duffy, is that in practice the pitch of the rivets must be divided up so as to come out even in the given length of plate, else one section will be weaker than the rest. To do this it's best to measure the length and bring the odd pitch near the girth seam for, at this point, the shell is stiffened and strengthened by the hoop or girth-seam lap. In the girth seam the total lap should be three rivet-hole diameters. The gain in strength is then one diameter, or 33 per cent., at the girth as regards the cylinder, at this point.——An' there ye are, Duffy, in laying out butt-strap joints, the City Hall bunch won't have annything on ye if ye get this soaked into your system even if ye be only a sub for a Chink."

# Energy Drop in Steam Turbines*

By E. Cardullo

*The effect of fluid friction is to increase the total heat and the entropy of the steam as it flows through the turbine and to increase the proportion of power developed in the later stages. An empirical formula is proposed for estimating the quantity of power developed in each stage of a turbine.*

*Abstract of paper delivered at the spring meeting of the American Society of Mechanical Engineers.

There are three general methods open to the designer for determining the properties of steam during its passage through the turbine. He may make use of empirical equations giving the relation between the heat content, entropy and temperature or pressure of expanding steam, as was suggested by Doctor Steinmetz.† He may make use of Mollier's total heat-entropy diagram which gives the relation between the total heat, entropy, pressure and quality of steam. Or he may make use of a table like that of Professor Peabody's giving the relation between the temperature, entropy, total heat, quality and specific volume of steam. The last two methods are more simple and more accurate than the first one and are to be preferred.

Assume that a turbine is to be designed having $n$ stages and that the diameters of the moving elements of each stage are the same. The heat drop per stage will be $\frac{1}{n}$ of the total heat drop. Were there no retransformation of work into heat, it would be necessary only to find from an entropy table or diagram the entropy and total heat of the steam as it enters the turbine, and the total heat, at the same entropy, of steam of the terminal pressure, to subtract the second quantity from the first in order to obtain the total heat drop, and then to divide this drop by the number of stages to obtain the heat drop per stage. The pressures in each stage would then be found by subtracting the heat drop per stage $n$ times from the initial heat content and finding from the table or diagram the pressure of steam having the heat content so found, at the given entropy. This method may be illustrated by the following problem:

Assume that the initial steam pressure is 164.8 pounds per square inch and the final pressure is 1.005 pounds per square inch; that the steam is initially dry and saturated, and that the number of stages is two. From Professor Peabody's table, the initial entropy is found to be 1.56 and the initial heat content 1193.3 B.t.u. The heat content of steam of 1.56 entropy at the terminal pressure is 871.1 B.t.u. The difference between the initial and final heat content, or the heat drop, is 322.2 B.t.u. The heat drop per stage is one-half this or 161.1 B.t.u. The heat content of the steam entering the second stage is

$$1193.3 - 161.1 = 1032.2 \ B.t.u.$$

The pressure of steam having this heat content and the entropy 1.56 is 18.4 pounds, which would be the absolute pressure of the steam as it enters the steam chest of the second stage.

†*Proceedings*, American Society of Mechanical Engineers, March, 1908.

In the actual steam turbine, however, the quantity of heat transformed into work is 40 to 70 per cent. of the heat theoretically available for transformation by isentropic expansion. Most of the missing energy has been retransformed into heat by eddying, fluid friction, blade leakage, etc., and appears in the steam, increasing its entropy. Assume that in actual practice 60 per cent. of the energy theoretically developed in the first stage of this turbine, or 96 B.t.u., would be transferred to the rotating member, and about 40 per cent., or 64.5 B.t.u., would be retransformed into heat, making the heat content of the steam entering the second stage,

$$1193.3 - 96.6 = 1096.7 \ B.t.u.$$

This would give for the entropy of the steam at the pressure of 18.4 pounds, the value 1.655. The heat content of steam of 1.655 entropy and 1.005 pounds pressure is 925 B.t.u.. which gives for the heat drop in the second stage

$$1096.7 - 925 = 171.1 \ B.t.u.$$

This is more than 6 per cent. greater than the heat drop assumed for the first stage. It is plain that in order to equalize the heat drop in the two stages, the pressure range in the first stage must be increased at the expense of that in the second stage.

It will be found by trial and adjustment that if the theoretical heat drop per stage, which may be designated by the symbol $\frac{\Delta H}{n}$, be multiplied by the empirical factor $(1 + K)$ the result will be the heat drop per stage which will give an actual equality in the quantity of energy developed in each stage. The value of $K$ is found by the equation.

$$K = 0.00056\left(\frac{n-1}{n}\right)\Delta H\,(1 - E)$$

where $n$ is the number of stages, $\Delta H$ the total heat drop theoretically available by adiabatic expansion between the initial and terminal pressures, and $E$ the probable thermal efficiency of the turbine. The value of $E$ may be obtained from the equation

$$E = \frac{2545}{S \times \Delta H}$$

$S$ being the probable steam consumption per horsepower-hour of the turbine.

In the case under consideration this efficiency has been assumed as 0.60, the heat drop as 322.2 B.t.u. and the number of stages to be two.

Substituting these values,

$$K = 0.00056\left(\frac{2-1}{2}\right)322.2\,(1 - 0.60)$$
$$= 0.036$$

The probable heat drop per stage will therefore be

$$\frac{322.2}{2}(1 + 0.036) = 167 \ B.t.u.$$

Allowing this drop in the first stage, the heat content after the first isentropic expansion will be

$$1193.3 - 167 = 1026.3 \ B.t.u.$$

From the entropy table, the pressure of the steam entering the second stage will be 16.86 pounds, since this is the pressure corresponding to the entropy 1.56 and the heat content 1026.3. Since the efficiency of the turbine is 60 per cent., the heat transformed into work is 60 per cent. of the theoretical heat drop and the heat transformed into work per stage is

$$\frac{\Delta H}{n} \times E = \frac{322.2}{2} \times 0.60 = 96.7 \ B.t.u.$$

Subtracting this quantity from the initial heat content, the heat content of the steam entering the second stage of the turbine is found to be

$$1193.3 - 96.7 = 1096.6 \ B.t.u.$$

Hence, the entropy of the steam entering the second stage is 1.663. Assuming the heat drop in the second stage to be the same as that in the first stage, the steam leaving the second set of nozzles will contain

$$1096.6 - 167 = 929.6 \ B.t.u.$$

The pressure of steam having 929.6 B.t.u. heat content at the entropy 1.663 is found to be 1.005 pounds, which gives a complete check on the work and shows the calculations to be correct.

If it is desired to find only the pressure of the steam as it enters each stage of the turbine, the following procedure may be employed: From the initial heat content of the steam $H_1$, subtract the quantity

$$\frac{\Delta H}{n}\left[1 + 0.00056\left(\frac{n-1}{n}\right)\Delta H\,(1 - E)\right] = h_1 \qquad (1)$$

and write

$$H_1 - h_1 = H_2 \qquad (2)$$

From the temperature-entropy table determine the pressure of steam of the initial entropy, having for its heat content $H_2$, which will be the pressure of the steam entering the second stage. Now subtract from $H_2$ the quantity

$$\frac{\Delta H}{n}\left[1 + 0.00056\left(\frac{n-3}{n}\right)\Delta H\,(1-E)\right] = h_2 \qquad (3)$$

and obtain

$$H_2 - h_2 = H_3 \qquad (4)$$

The heat content, $H_3$, together with the initial entropy of the steam, will determine the pressure of the steam entering the third stage. The pressure of the steam entering the fourth, fifth, etc., stages is obtained in a similar manner except that the quantities $\frac{n-5}{n}$, $\frac{n-7}{n}$, etc., must be substituted for the quantity $\frac{n-1}{n}$ in (1) to obtain the quantities $h_3$, $h_4$, etc. When performing this operation in the case of any particular turbine, it will be found that the value of $h$ is greater than $\frac{\Delta H}{n}$ for the $\frac{n}{2}$ high-pressure stages, and less than $\frac{\Delta H}{n}$ for the $\frac{n}{2}$ low-pressure stages.

## Pressure Temperature Relations of Saturated Steam *

By L. S. Marks

Relations between the pressures and temperatures of saturated steam are accurately known for temperature ranges of 32 to 400 degrees Fahrenheit. Within this range the experimental values of Regnault and other investigators agree very closely with the recent work of Scheel and Heuse and of Holborn and Henning. At higher temperatures and pressures, however, there is no such agreement between the results of the different investigators. Of the later investigations within this higher range, the results of Holborn and Baumann appear to be the most authoritative. These covered a range between 400 degrees Fahrenheit and the critical temperature, indications pointing that the latter condition is reached at 706.3 degrees Fahrenheit with a corresponding pressure of 3200 pounds per square inch.

The measurement of vapor pressure may be by either the statical or the dynamical method. In the statical method the liquid and its vapor are maintained at a constant temperature and the corresponding pressure is measured. In the dynamical method the pressure is kept constant and the corresponding temperature is measured. The pressure is maintained by air or gas acting on top of the liquid, which is heated continuously, and the vapor which forms is condensed and returned by gravity.

The work by Holborn and Henning on saturation temperatures from 120 to 400 degrees Fahrenheit was by the dynamical method, and that of Holborn and Baumann by the statical method. As carried on by the latter investigators, water was contained in a steel vessel surrounded by a constant-temperature bath; absolute measurements of the pressure were obtained by means of a weighted rotating plunger; and the volume of water could be varied either continuously at an approximately uniform rate, or intermittently. It was noted that the water acted on the walls of the steel vessel and that after repeated heating to over 570 degrees, a small quantity of iron went into solution, the water becoming discolored upon standing exposed to the air.

Table 1 gives the results of Holborn and Baumann as compared with those of Cailletet and Colardeau and those computed by the formula

$$log.\ p = 10.15354 - 4873.71\ T^{-1} - 0.00405096\ T + 0.000001392964\ T^2$$

where $p$ is the absolute saturation pressure corresponding to the absolute temperature $T$.

*Abstract of a paper read at the spring meeting of the American Society of Mechanical Engineers.

TABLE 1. EXPERIMENTAL AND CALCULATED PRESSURES OF SATURATED STEAM FROM 400 DEGREES FAHRENHEIT TO THE CRITICAL TEMPERATURE

| Temperatures, Degrees Fahrenheit | Pressures, Pounds per Square Inch | | | Deviation of Formula from Holborn and Baumann | |
|---|---|---|---|---|---|
| | Cailletet and Colardeau | Holborn and Baumann | By Formula | Pounds per Square Inch | Percentage |
| 400 | 247.1 | 246.99 | 247.10 | +0.11 | 0.044 |
| 410 | 276.4 | 276.34 | 276.47 | +0.13 | 0.047 |
| 420 | 308.4 | 308.33 | 308.47 | +0.14 | 0.045 |
| 430 | 343.2 | 343.18 | 343.26 | +0.08 | 0.023 |
| 440 | 380.8 | 380.92 | 381.02 | +0.10 | 0.026 |
| 450 | 421 | 421.85 | 421.87 | +0.02 | 0.0047 |
| 460 | 465 | 465.95 | 466.04 | +0.09 | 0.019 |
| 470 | 573 | 513.65 | 513.66 | +0.01 | 0.0019 |
| 480 | 565 | 565.08 | 564.93 | —0.15 | —0.026 |
| 490 | 622 | 620.18 | 620.05 | —0.13 | —0.021 |
| 500 | 684 | 679.26 | 679.18 | —0.08 | —0.012 |
| 510 | 751 | 742.55 | 742.56 | +0.01 | 0.0013 |
| 520 | 822 | 810.31 | 810.37 | +0.06 | 0.0074 |
| 530 | 897 | 882.58 | 882.82 | +0.24 | 0.027 |
| 540 | 977 | 959.85 | 960.15 | +0.30 | 0.031 |
| 550 | 1062 | 1042.2 | 1042.6 | +0.4 | 0.038 |
| 560 | 1152 | 1130.2 | 1130.3 | +0.1 | 0.0089 |
| 570 | 1247 | 1223.7 | 1223.7 | 0 | 0 |
| 580 | 1349 | 1323.0 | 1322.9 | —0.1 | —0.0076 |
| 590 | 1458 | 1428.3 | 1428.1 | —0.2 | —0.14 |
| 600 | 1574 | 1539.9 | 1539.8 | —0.1 | —0.0065 |
| 610 | 1697 | 1657.8 | 1658.1 | +0.3 | 0.018 |
| 620 | 1827 | 1782.9 | 1783.3 | +0.4 | 0.022 |
| 630 | 1965 | 1915.3 | 1915.9 | +0.6 | 0.031 |
| 640 | 2111 | 2055.1 | 2056.0 | +0.9 | 0.044 |
| 650 | 2265 | 2203.1 | 2204.2 | +1.1 | 0.049 |
| 660 | 2482 | 2359.2 | 2360.5 | +1.3 | 0.055 |
| 670 | 2599 | 2523.4 | 2525.6 | +2.2 | 0.067 |
| 680 | | 2697.1 | 2699.7 | +2.6 | 0.096 |
| 690 | | 2882.3 | 2883.3 | +1.0 | 0.035 |
| 700 | | 3080.4 | 3076.8 | —3.6 | —0.117 |
| 706.1 | | 3200.0 | 3200.0 | 0 | 0 |

TABLE 2. EXPERIMENTAL AND CALCULATED PRESSURES OF SATURATED STEAM FROM 32 TO 400 DEGREES FAHRENHEIT

| Temperatures, Degrees Fahrenheit | Pressure, Pounds per Square Inch | | Deviations of Formula from Tabulated Values | |
|---|---|---|---|---|
| | From Marks & Davis' Steam Tables | By Formula | Pounds per Square Inch | Percentage |
| 32 | 0.0886 | 0.088563 | —0.000037 | —0.042 |
| 50 | 0.1780 | 0.17765 | —0.00035 | —0.196 |
| 100 | 0.946 | 0.946 | 0 | 0 |
| 150 | 3.714 | 3.707 | —0.007 | —0.188 |
| 200 | 11.52 | 11.504 | —0.016 | —0.139 |
| 250 | 29.82 | 29.802 | —0.018 | —0.060 |
| 300 | 67.00 | 67.00 | 0 | 0 |
| 350 | 134.6 | 134.60 | 0 | 0 |
| 400 | 247.1 | 247.10 | 0 | 0 |

The agreement of the pressures calculated from this equation with the experimental results of Holborn and Baumann is very striking; from 400 to 650 degrees the difference is less than $\frac{1}{20}$ of 1 per cent., and from 650 degrees to the critical temperature the maximum difference is slightly over $\frac{1}{10}$ of 1 per cent.

Below 400 degrees Fahrenheit, the agreement of this equation with the experimental results is shown by Table 2. The experimental results are those of Holborn and Henning from 120 to 400 degrees Fahrenheit and of Regnault and other investigators for temperatures below 120 degrees. The differences are very small, as expressed in pounds per square inch, but amount to nearly $\frac{1}{5}$ of 1 per cent. in some cases. This, however, is not greater than the variations among the best experimental values in this part of the temperature range.

According to a recent French patent, an aluminum solder may be made by first making a fusible alloy—which will melt in boiling water—of three parts of tin, eight parts of bismuth, and five parts of lead. The solder itself is then made by taking ten parts of the fusible alloy, 300 parts of zinc and five parts of aluminum. This is said to make a strong solder. A softer one is made by taking 160 parts of the fusible alloy, 80 parts of zinc, 25 parts of aluminum and 80 parts of tin.

## An Old Time Tide Mill

When an engineer uses Slade's mustard or other seasonings for his meat at dinner, he does not stop to think that the power to grind out these delicacies was furnished by an old-time tide mill. In 1800 the legislature passed a bill which permitted the damming of the Snake river, which runs through the towns of Chelsea and Revere. The mill building burned down twice. The present structure and a view of the tide wheels and the water escaping from them to the river are shown in Fig. 1.

FIG. 1. SLADE'S TIDE MILL, SHOWING WATER FLOWING FROM THE WHEELS AT LOW TIDE

The tide wheels are four in number and generate a total of 100 horsepower. One wheel is 40 inches in diameter, and three are 54 inches in diameter. These wheels are operated six hours on each tide, and a day and night crew are kept at work. When the mill is not in operation, the men are taking care of the material which has been ground; thus there is no loss of time on the part of the workmen.

In the dam a raceway has been provided, at one end of which two gates are hung, to swing from one end, so that as the tide comes in through the raceway the flow automatically opens these gates, which remain open until the turn of the tide, when the flow of water swings them closed. The average flow of tide at this point is about 10 feet, but it is not necessary for the tide on the sea side of the gates to fall below the level of the wheels before the mill can be put into operation. The mill, therefore, can be operated as soon as there is a sufficient difference in head of the water on the two sides of the dam.

The flow of water is controlled by ordinary flat gates which allow water to flow from the pond side of the dam through four penstocks to the turbine waterwheels.

Fig. 2 shows two of the waterwheels under the mill. Recently one of the wheels required renewing, and upon looking up the records it was found that it had been in operation for a period of 25 years before the action of the salt water had deteriorated it. The cost of repairs and renewal amounts to practically nothing.

The original spice mill was bought in 1847 by two Slade brothers, and they were one of the pioneer companies to introduce ground spice to the trade. This mill is now the property of the D. & L. Slade Company, Revere, Mass.

## Water Jacket Deductions

BY FRANK RICHARDS

Some interesting facts were recently brought out in a report by A. Hoffman regarding a test of a compound steam and two-stage compressor at the shops of the Ingersoll-Rand Company. This compressor had 20- and 38-inch steam cylinders, 33¼- and 20¼-inch air cylinders and a common stroke of 30 inches, the capacity being 3000 cubic feet of free air per minute.

Much of the material in the report is in the nature of "company data," but through the courtesy of the company, I am able to give extracts concerning the water jacket and the method of accounting for the heat thereby abstracted. What follows is taken almost verbatim from the report.

"It is well known from the principles of thermodynamics that the foot-pounds of work done in the air cylinder during any period of time, is equal to the product of the increase in temperature produced by compressing the air, the weight of the air, the specific heat of the air and Joule's equivalent (778). In making up the heat balance this was found to check closely with the results found in practice."

"The work done in the cylinder must be equal to the heat given to the air-cylinder water jacket plus the heat contained in the air after leaving the cylinder, above the temperature of the intake air. Therefore, the heat given to the jacket is 2070 B.t.u. per minute, which is equivalent to 48.8 horsepower. The heat contained in the air above the intake temperature of 77 degrees is 8320 B.t.u. per minute, and this is equivalent to 196.3 horsepower. The total B.t.u. is the sum of these two quantities, or 10,390 B.t.u. which is equivalent to 245 horsepower. Theoretically, this total should represent the actual horsepower indicated in the low-pressure air cylinder, and it was found that this horsepower was 243; the difference being only 2.1 horsepower. A greater difference than this might be expected because the actual amount of air was figured from the vol-

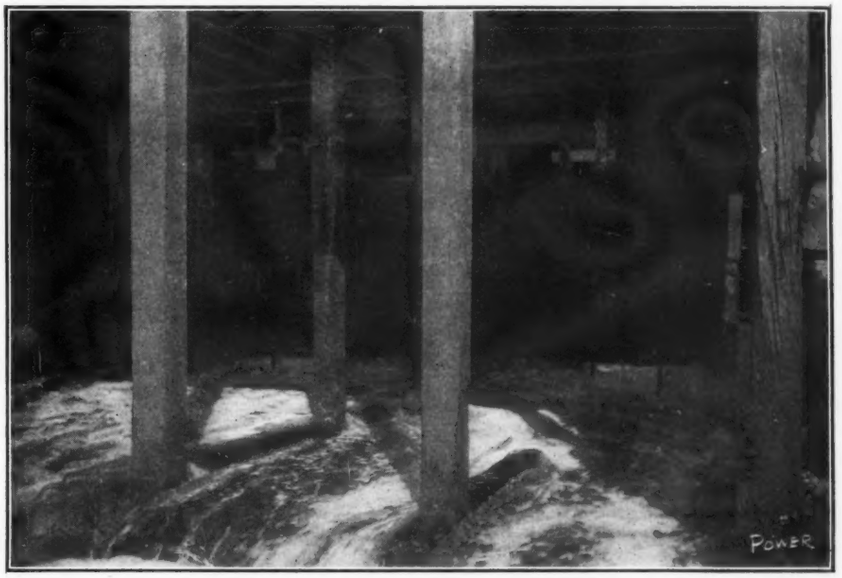



FIG. 2. NEAR VIEW OF TWO OF THE WATER WHEELS

umetric efficiency of the indicator card, which, of course, was not absolutely correct."

"There was also a discrepancy due to the radiation of the outside air to the water jacket, but this was very slight, as was shown by the fact that the temperature of the water leaving the jacket before the compressor was started was the same as the temperature of the cold water in the main."

"It was at first thought that there was something wrong regarding these results, as they showed about 20 per cent. of the total indicated work in the cylinder to be given up to the jacket water; according to this considerably better than isothermal compression should be obtained, which, of course, would be impossible. The explanation is as follows."

"Very little heat is given to the water jacket while the air is compressed, because the compression begins at a low temperature, and the maximum temperature is not reached until the end of compression, and while at the maximum temperature, the piston is traveling very fast and there is not much chance for heat to be given up. After the discharge valves open, however, a great deal of heat is given to the jackets because during this period the air is at its maximum temperature and it also comes in intimate contact with the jacket of the air-cylinder head in passing out through the valves; in addition to this, the piston is traveling at a comparatively slow speed toward the end of the stroke. Some heat is also given to the jacket while the air is passing out through the discharge passage."

"This explanation is sufficient to account for the large amount of heat given to the jacket, and it shows that jackets really do more good than is usually supposed. Of course, heat given to the cylinder jacket while the air is discharging does not reduce the work in the cylinder but merely lowers the temperature of the air and raises the temperature of the jacket water."

# Inertia Effects and Shaft Couplings

By H. J. Smith

*The various factors entering into the choice of a suitable clutch are considered and a table giving horsepower ratings and equivalent shaft diameters is presented.*

The influence of inertia in the starting of heavy masses into motion is well understood in installations using electric power. An electric motor under such conditions will show disturbances at the commutator, and the abnormal current consumption can be plainly noted at the switchboard meters. Special methods of winding field and armature coils, together with the introduction of resistance or controlling devices, are made to enable the motor to pick up the load of a trolley car, elevator, or other heavy starting loads. Such installations of motors, when of sufficient magnitude, are well studied, the data are reasonably exact and the results satisfactory. But with the occasional installation, where the data are not definite, the possibility of poor operative results insures a careful investigation and a recommendation from the builder of the motor.

In ordinary power-transmission work, either by belt, rope, gear, shaft or clutch drive, little thought is usually given to inertia. Horsepower capacities of driven machines are either assumed or obtained from the catalogs of machine manufacturers. These are roughly taken as a basis, to which may be added some additional values dictated by experience, or, as ofttimes happens, simply an extra haphazard allowance to increase the factor of safety.

In general, to engineers and those conversant with power-transmission machinery, belts, ropes, gears and shafting are easily figured for a given duty because the strength of material is well known, and with a proper choice of cross-section a factor of safety can be taken which will be ample for even extraordinary overloads. Speed merely increased the capacities of any of these transmission members, and so long as the most efficient speeds are maintained or the safe limits not exceeded, speed is not a disturbing factor in the calculations. If the transmitting machinery is to start with the engine or motor, the inertia loading will be overcome gradually, provided the limits of safety of the material are not exceeded by the stresses developed. If there is no breakage or slippage, then as the speed increases the inertia stresses decrease, until at the regular speed all parts are under the least stress. The duty thereafter imposed on the parts varies only according to subsequent work performed.

HORSEPOWER RATINGS AND EQUIVALENT SHAFT DIAMETERS

| Horsepower at 100 R.p.m. | Equivalent Shaft Diameters |
|---|---|
| 9 | 1 3/16 |
| 12 | 1 1/16 |
| 15 | 2 3/16 |
| 20 | 2 5/16 |
| 27 | 2 7/16 |
| 35 | 2 9/16 |
| 45 | 3 5/16 |
| 60 | 3 7/16 |
| 75 | 3 11/16 |
| 90 | 3 13/16 |
| 110 | 4 1/16 |
| 140 | 4 11/16 |
| 175 | 4 15/16 |
| 230 | 5 1/2 |
| 350 | 6 |
| 480 | 7 |
| 625 | 7 1/2 |
| 875 | 8 1/2 |
| 1300 | 10 |

While these calculations are relatively simple and reasonably effective for the transmission machinery mentioned, there are no such reliable formulas in the choice of a friction-shaft coupling. The latter is a great convenience not only because of the flexibility attained in the system whereby a part (or parts) may be operated at will but because it stands between the work to be done and the engine, waterwheel or motor, and relieves the latter from heavy starting stresses. A gas engine in power-transmission work must either be under a very light inertia load at starting or be started with no load at all. With most gas engines a clutch coupling is desirable. The waterwheel receives a full draft of water from the steam, and the motor its increased current from the generator. All that is required of either are strength and endurance to meet the shock or strains of starting and carrying loads to their ultimate capacity with continuous motion; variations of load are met by governing devices.

In the transmission of energy through a friction-clutch coupling, there is no automatic device to meet overload conditions. The clutch must rely on a fixed pressure which can only be regulated when the mechanism is at rest, while every time it is engaged it losses some of this pressure through wear at the friction surfaces and thereby has its capacity reduced; yet to operate successfully the clutch coupling is expected to practically take care of itself, and to be always reliable and not the cause of shutdown.

In the usual application of a friction-clutch coupling the power end of the transmission machinery has already been brought up to normal speed and is operating under the governor, and ready for any load within maximum limits. The load end is at rest. The friction-clutch coupling must then stand the brunt of the starting load. Half of it at this moment is at speed while the other half is at rest. The full shock, therefore, of starting a great mass at rest into motion, overcoming the total friction of rest, the distortion of parts, and often in addition a part or even the whole of the maximum load resistance, is required of the friction-clutch coupling. The

whole work must be accomplished by the friction surfaces of the clutch which can only be under partial clamping pressure to allow a constant slip until both members of the clutch coupling are at the same speed. This slip, which may be of short duration for light inertia loads or extend over a considerable period of time in starting heavy loads, is the one indefinite factor in any calculation involving friction-clutch couplings. While under pressure a slip of the friction surfaces (which in power-transmission mechanism are usually hard dry maple on cast iron) produces abrasion and heat which rapidly remove the wood in proportion to the amount of work done and to the length of time the slippage continues. If either or both are very great, then it is quite possible that the wood surface may be so worn away that the clutch, while able to start the mass into motion, may fail to bring it up to speed, or if it does, may slip when a later loading is placed upon it. Unlike other mentioned power-transmission members, the question of sufficient strength to transmit a given horsepower is not the main consideration. A clutch can be made large and strong enough in its parts to do any stated work; but it must also have the prime requisites of large friction surfaces to resist wear and an easy means of renewing or readjusting the friction surfaces. The pertinent questions then in determining the size of a friction-clutch coupling are:

First, the normal running load.

Second, starting inertia load.

Third, the speed of either member of the clutch coupling at the time of engagement.

Fourth, the number of engagements in a given time.

Fifth, the diameter of the shaft (or shafts) on which the coupling or either of its members is mounted.

Considering these headings in brief detail it is fairly easy to fix on the normal running load of a machine, although its capacity to consume power is often underrated. The total power for more than one machine is not exactly proportional to their number, but is something less, depending upon the inertia stored up in the revolving parts of the machines and transmission machinery. This inertia, or flywheel effect, equalizes the fluctuation of delivered power, and therefore is of direct assistance to a clutch coupling when in motion.

In starting, however, the flywheel effect is entirely potential or even negative and resists the action of the clutch coupling to set the mass in motion. If this resistance is slight, then the clutch can be engaged in a minimum time and there will be very little wear of the friction surfaces. If the resistance is more severe, then the time of clutch engagement is protracted, the wear is great and the necessity for frequent adjustment of friction surfaces is essential. If the resistance to starting is absolute, then the duration of clutch engagement is simply limited by the total loss of clamping pressure of the friction surfaces occasioned by the actual burning of the wood shoes.

From the last paragraph it is obvious that in the choice of a proper clutch coupling the number of engagements is an important factor, and directly proportional to the severity of the service.

Recognizing the fact that the size of shaft chosen is an index to the character of the work done, manufacturers of clutch couplings have given an "equivalent shaft" rating to each size of clutch listed, based on a standard of 100 revolutions a minute and a reasonable factor of safety for both. While this approximate proportioning does fairly well in practice, it must not be assumed that the clutch is really of equivalent strength with the shaft under all conditions. A homogeneous steel shaft has a greater factor of safety in its ability to resist torsion than the cast-iron clutch, while the latter, because of its inelasticity, is still less capable of withstanding shock. Therefore the "equivalent shaft" rating of cast-iron clutch couplings is simply a convenient guide to prevent underestimating the size of the clutch coupling. There are usually more reasons to make the clutch larger and very few to make the size of clutch coupling smaller than in the accompanying table.

The ratings of clutches when based on actual prony-brake tests at a speed of 100 revolutions per minute show a considerable factor of safety as far as actual strength is concerned but the factor of endurance without readjustment is much less. Within usual limits of speed each clutch can easily start its rated load from rest with a sufficient factor of endurance. In practice the ordinary requirements for starting capacity are less than for the working-load capacity, but there are very many installations in which the inertia load at starting exceeds the rated capacity of the clutch. A clogging of machines, gears that bottom, abnormally tight belts, the attempt to too quickly accelerate heavy masses, produce extraordinary strains and decrease the endurance of the clutch.

The installation that gives trouble sometimes results from insufficient information in the hands of the clutch builder, but more often follows inaccurate deductions of the purchases on catalog ratings, speed inertia effects and price. Poor judgment may be some excuse, but to deliberately place too small a mechanism because of its cheapness is the poorest of all excuses for an unsatisfactory installation.

As a concrete example of very severe inertia loading consider the following application of a clutch coupling:

A 400-horsepower gas engine is belted to a 5 $\frac{7}{16}$-inch jack shaft and drives the latter 450 revolutions per minute. The 4 $\frac{15}{16}$-inch mill shaft extended along the line of the jack shaft and had to be connected by a cutoff friction-clutch coupling so that the gas engine could be started light and the mill afterward started by means of the friction coupling. The character of the load was severe, and consisted principally of a battery of large attrition mills. These mills, on account of their construction, high speed and liability in service to be in a more or less clogged condition in starting, offer inertia-loading conditions almost impossible to calculate. Therefore, the first step was to predicate the size of the clutch coupling on the diameter of the driven shaft already fixed by the experience of the mill man. From the table, a 175-horsepower clutch coupling is the minimum size, but a larger size is preferable on account of the shock load and the larger friction surfaces. However, even a 175-horsepower clutch coupling could not be used because of the peripheral speed (4950 feet) of its friction surfaces exceeding the maximum safe speed of 3600 feet per minute (beyond this speed, wood friction surfaces char rapidly when the clutch is engaged), and secondarily because the outer periphery of the clutch exceeded the safe speed limit of cast iron—a mile a minute. For these reasons a 140-horsepower clutch was chosen which reduced the friction-surface peripheral speed to 4500 feet per minute.

The severe conditions in this installation point the moral of inertia loading. The 140-horsepower clutch (rated at 100 revolutions per minute), which appears large enough in view of the speed of 450 revolutions per minute, actually handled the running load without the least symptom of trouble; but the heavy inertia load at starting, aggravated by the high speed of the driving-clutch member, wore the wood shoes so fast that the assistance of a mill hand pulling on the belts was necessary to start the mill and at the same time to keep the friction surfaces in good condition for subsequent loads without constant readjustment. A slower speed of mill shaft, if chosen in the first place, would have allowed the choice of a clutch of ample size for the work with a peripheral speed well within safe limits for wear of friction surfaces.

---

A locomotive was run into the roundhouse after its first trip since having been overhauled and new flues put into the boiler. The foreman asked the darkey fireman how it steamed, and how much pressure he was able to keep on it. The darkey replied: "Ah don't know jes how much steam Ah had on 'er, but Ah sho did keep th' needle pintin' ovah towge th' bilah."

# Electrical Department

*Especially conducted to be of interest and service to the men in charge of the electrical equipment*

## Air Cooled Choke Coils

It is sometimes convenient to mount choke coils on ceilings so that the insulating-coil supports hang pendent and at other times it is preferable to arrange them so that the coil rests on the supports, as shown in the engraving. The new Westinghouse choke coil, which this engraving illustrates, can be mounted either way, because the insulating "petticoat" columns can be removed and inverted on the supporting rods.

The choke coil consists of an aluminum rod bent into a helix of about 15 inches diameter and containing about 30 turns. Bracing clamps are provided to give mechanical strength to the helix. The aluminum rod used is of sufficient cross-section to safely carry 200 amperes.

AIR-COOLED CHOKE COIL

Each of the two insulating columns is made up of porcelain insulators which, except the end pieces, are interchangeable. The number of insulators used in the column depends on the voltage of the circuit in which the coil is to be used. For floor mounting, the parts are arranged as shown here.

These choke coils are intended principally for the protection of transformers and should not be used for generators. Where greater reactance than is afforded by a single coil is desired for the higher voltage circuits, it is recommended that two or more coils be connected in series.

## Factors in Good Service on Transmission Systems*

BY M. HILGEN

Absolutely uninterrupted service from a transmission system is impossible. Good and commercially acceptable service can be obtained, however, from any system which is properly designed, constructed and operated. For maintaining continuity of service more can be done before the system is constructed than at any time thereafter.

The most important factors in securing good service are insulation, mechanical strength, general design of the system and methods of operation.

### INSULATION

Probably more line shutdowns have been produced by insulator failures than by any other cause. The general reason is that, although other engineering structures are ordinarily built with factors of safety ranging from 4 to 10, insulators, for some unaccountable reason, have been considered adequate if they had a factor of safety of 2 or possibly 2½, notwithstanding the fact that the stresses which insulators are called upon to stand are probably more variable and uncertain than those encountered by most other structures. The higher the line voltage the less the factor of safety can be. For a 100,000-volt system a factor of safety of 3 under unfavorable conditions (with the insulator wet all over) gives excellent results. For 50,000 volts the factor of safety should be 4 and for 20,000 or 30,000 volts I believe a factor of safety of 5 is about right. The reason for this is that insulators rarely fail due to the normal voltage of the system, but rather to abnormal voltages caused by lightning and power surges; these abnormal voltages are greater in proportion to the normal voltage for a low-tension than for a high-tension system.

To obtain a proper factor of safety for a 50,000- or 60,000-volt system with pin insulators is difficult; for a 100,000-volt system it is practically impossible. This fact led to the design of the suspension insulator, the introduction of which marked an important advance in high-voltage line construction. For voltages of 20,000 and above, no other insulator should be used.

### MECHANICAL STRENGTH

For straight construction in level country, the stresses to which a line will be subjected can be predetermined with a reasonable degree of accuracy and proper factors of safety allowed. For extra long spans or sharp angles the stresses can also be determined in advance and special pole structures or towers can be constructed to meet the conditions.

Where a line traverses rough and mountainous country for many miles, however, and no two spans are of the same length and no two towers or poles are of the same hight, each span and tower should theoretically be treated as a special case, which is obviously impossible; general rules must be formulated for the construction of the line as a whole. Towers or poles must be selected which are sufficiently strong to stand in the most unfavorable places and these guyed, when necessary, as an extra precaution.

If wire larger than No. 6 Brown & Sharpe gage is used it should be stranded. For No. 4 a three-strand wire gives excellent results, the strands being so large that no individual wire is liable to be broken by abrasion. For sizes of from No. 2 to No. 0000, seven-strand wire should be used. Six-strand wire with a hemp center exposes more surface to wind and ice, is less flexible and tends to crush out of shape in the wire clamps.

Hard-drawn copper wire should not be drawn too hard with the idea of getting great tensile strength. Wire having an ultimate strength of 45,000 to 50,000 pounds per square inch and an elastic limit of 26,000 pounds per square inch is strong enough. Stronger and harder wire than this is liable to become nicked or scratched in stringing or injured by the wire clamps so that the actual strength of the wire when erected will be less than the actual strength of a softer wire.

Wire clamps for suspension insulators should be connected to the insulator by

*Extracts from a communication presented at the New York convention of the National Electric Light Association.

a hinge joint located as near as possible to the wire so that there will be little tendency to turn the body of the clamp and bend the wire when the insulator pulls in a direction other than perpendicular to the wire.

The insulators should be hung so that they can swing freely in any direction. If the connecting device does not allow perfect freedom of motion it is liable to be broken.

### General Design of System

Transmission lines feeding an important load center should be in duplicate or triplicate. If they come from the same power plant they should preferably follow different routes. If this is impracticable on account of the extra cost of right-of-way and patrol, the different lines should at least be mounted on separate towers or poles, spaced far enough apart to prevent the possibility of one line interfering with another.

For convenience in locating trouble, all lines should be sectionalized by means of outdoor air-break switches, located about twenty miles apart and at patrolmen's headquarters if possible. Two long duplicate lines should be provided also with a cross-over switching station at the middle of the line with the switches so arranged that one-half of either line may be cut out and the remaining section continued in operation. This switching station may be an outdoor affair with air-break switches, or if the charging current of half of one line cannot be broken by air-break switches, oil-break switches may be used and installed indoors or out.

If the load center is supplied by two or more separate and independent plants the conditions are better, because the chances of two independent supply systems being out of commission at the same time are remote.

In any case where two or more lines supply power to the same point, the incoming lines should be controlled by reverse-current relays. The greater the number of incoming lines the better the relays will work. Grounds rather than short-circuits between wires cause the great majority of line troubles. On a Y-connected system with a grounded neutral, any other ground causes a short-circuit and takes a considerable amount of power. For this reason reverse-current relays are of greater service on a Y-connected system than on a delta-connected system. On a very extensive delta-connected system, however, one having large line capacity, a grounded wire will take a heavy current and this current, on account of the resistance of the ground and the resistance of the wire, will represent true power. A very large system with an extended network and many plants will work equally well whether Y- or delta-connected.

Trouble from lightning can be guarded against by high insulation, one or more overhead ground wires and electrolytic arresters; these represent the highest development yet attained in lightning arrester construction. The ground wires should be of ample section and grounded at every support.

### Operation and Supervision

In the operation of a transmission system eternal vigilance is the price of continuous service. No amount of automatic apparatus can take the place of good operators.

New operators should first be placed in the smaller and less important stations or substations of the system; as they prove their ability and as vacancies occur, they should be transferred to the larger and more important stations. Under this plan the men have always something to look forward to and do not lose interest in their work. Only the best men get to be operators in the more important stations, and the men in the larger stations have necessarily had considerable experience and are well acquainted not only with the operation of their own station but with the switching connections and method of operation of several of the other stations of the system.

In very large systems all operations should be controlled by one man, called the "load despatcher." He should preferably be located at the principal load center and be in communication with all stations through a private telephone system as well as the public systems where one is available. In smaller systems the load despatcher may also be the operator in charge of one of the large stations.

The frequency with which a line should be patrolled depends upon the importance of the line and its liability to failure. An old wooden-pole line with pin insulators may have to be patrolled every other day if it supplies power to an important load center. A well constructed steel-tower line may need to be patrolled only once a week or less.

Certain parts of a line may need closer attention than others, notably those in mountainous country and swamps. Unimportant branch lines may be patrolled once a month, or perhaps not at all. A branch line to a small power plant is less important than a branch line to a smalll substation.

Patrolmen should carry portable telephones and report their location to the load despatcher at regular intervals. The record of the movements of the patrolmen can conveniently be kept by the load despatcher on a "patrol sheet." This is a continuous roll of cross-section paper with a time scale on the base and a vertical distance scale. For convenience, all the stations, patrol houses and other landmarks are shown on the sheet according to their relative location along the line. The progress of each patrolman on the line is then noted by dots on the chart through which a curve is drawn, just as progress of a train is shown on a railroad-train sheet.

Patrolmen are too prone to think that their one object in life should be to cover a section of line at stated intervals and as expeditiously as possible. High-speed patrol is about as good as no patrol at all. In one case it was discovered that a certain patrolman had purchased a motor cycle and was patrolling his section of line at 30 to 40 miles per hour, following the adjacent highways rather than the line.

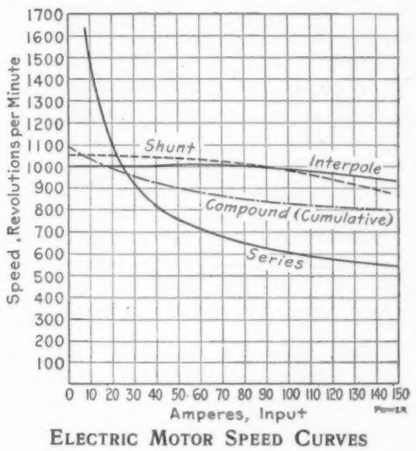
A slow and thorough examination of a line once a month with a sharp lookout to discover preliminary signs of weakness, such as loosening tie wires, cracked insulators, loose bolts in towers, broken wires in a strand, uneven sags in spans, settling of tower footings, loosening of guy wires, etc., will do more good than a high-speed inspection daily.

On a large system a general line inspector should be employed whose duty it is to inspect all lines, going from one to another at indefinite intervals and reporting all signs of weakness and making reports as to work of patrolmen and recommendations as to repairs, reconstruction and other necessary matters.

# CORRESPONDENCE

## Interpole Motor Characteristics

Referring to the discussion of Messrs. Wilbraham and Dean regarding interpole motors, appearing in the issue of May 23, I beg to take issue with both gentlemen as to some of their statements. Mr. Wilbraham's statements that the interpole winding adds no torque or counter electromotive force and that the speed curves of the shunt-wound motor and interpole motor are very similar, are incorrect. Although the interpole winding is added primarily to improve commutation and not with any idea of changing the speed, it does affect the counter electromotive force by tending to oppose the main field




Electric Motor Speed Curves

and this tends to increase the speed of the motor.

The speed curve of a shunt-wound motor always droops as the load increases. With an interpole winding the speed curve will not droop so much; it may rise and in some cases the speed may become dangerous when load is put on. For this reason it is the practice of motor builders to take a speed-characteristic test on all interpole motors. Motors having a rising speed curve are not passed for shipment until the trouble, due to too powerful an interpole field, is remedied, usually by shunting part of the current around the interpole winding.

I believe that Mr. Dean is right in saying that the trouble was due to wrong connection of the series winding, and I think if Mr. Wilbraham had reversed the series winding, leaving the interpole winding connected as at first, he would have had no trouble from sparking. From my own experience and from inquiries which I have made, I should say that a motor with a variable-speed ratio of more than 3 to 1 and with a differential compound winding is poorly designed and likely to run away when load is put on, as at the high speed the weakened shunt field would be easily overcome by the series field.

I can hardly agree with Mr. Dean's broad statement that "it is primarily an error to buy a compound-wound interpole motor" as there are undoubtedly cases where a differentially connected series field winding is desirable and safe for a constant-speed motor and there are other cases where a cumulative connection, giving a decided droop to the speed curve, is very desirable. Such cases, however, should usually be referred to the designing engineer.

The accompanying chart showing characteristic speed curves will help to explain my statements.

C. A. CALL.

Schenectady, N. Y.

## Central Station Service vs. Isolated Plant Operation

I have been very much interested in the various items appearing in POWER regarding the relative cost of isolated-plant operation and central-station power. Mr. Rushmore's article in the May 23 issue, to which I have referred in detail elsewhere in this issue, is but one example of the incompleteness of many of the reports made by the isolated-plant operators. On page 819 of the May 23 number, for example, A. P. Hyde states that his plant is producing electrical energy for 1.06 cents per kilowatt-hour and gives comparative coal and load data for three months of the year. If he gave a complete year's operation an intelligent comparison could be made, but the comparison as given is so incomplete as to be of little value.

Reliable and complete data of this kind would be of very great general interest and value. I believe that under certain conditions the isolated plant can produce its own power, heat and light at a lower figure than the total cost to the plant would be if taking central-station power and making low-pressure steam for heat. Under other conditions the reverse will be true. As these conditions of operation undoubtedly affect the result to such a very great extent, unusual care must be taken in accepting data, even when in complete form, and no snap judgment should be reached until the conditions have been thoroughly analyzed.

I should like to see the discussion of the relative value of the central-station service and the isolated-plant service continued even more thoroughly and in greater detail than it has been, but I would suggest that care be taken to see that no data are published which are not reliable and which do not consider all the elements of cost entering into the production of power.

R. D. DEWOLF,
Ass't Mech. Engr., Rochester Ry. and Light Company.

Rochester, N. Y.

## Parallel Operation of Alternators Driven by Waterwheels

I wish to submit to practical readers of POWER a problem that has recently occasioned considerable study on my part.

We have a 200-kilowatt 2300-volt three-phase alternator of the revolving-field type coupled to a hydraulic turbine, the speed of which is controlled by a Woodward governor. The exciter is belt-driven from the main shaft of the alternator.

On the switchboard there are three ammeters, one in each lead of the generator, and a voltmeter which can be connected across any two leads. There are also a voltmeter and ammeter for the exciter as well as a field rheostat and a field switch with discharge resistance.

This unit supplies current for both power and lights and it is quite up to its capacity. In order to increase the output it is proposed to install generators to be driven by two smaller waterwheels at adjacent water sites. There have been ordered one 60-kilowatt unit to be placed about a mile from the largest installation and a 36-kilowatt machine to be located a mile farther down stream.

The question at issue is this: Is it necessary to supply the small units with governors or will the one on the 200-kilowatt outfit be sufficient? We desire the two new units to work at maximum capacity constantly and thus relieve the large unit of a substantial portion of its load. What additional instruments will it be necessary or desirable to place on the 200-kilowatt switchboard and what are necessary at each of the two small power houses?

What is the proper sequence of operations in throwing either of the smaller machines on the line when either or both the others are in operation? And, in reverse order, how may one unit be shut down without disturbing the other two?

H. T. DEAN.

Boston, Mass.

## Starting Large Motor Generators

Sometimes it is found that a large motor-generator consisting of an interpole generator driven by a synchronous motor, when started from the direct-current side, takes excessive starting current. In such a case, if the generator is compound wound the best way to reduce the starting current is to reverse the connections of the series field winding so that the machine will operate as a compound motor, increasing the field excitation and thereby the starting torque per ampere of armature current. A double-pole double-throw switch can be easily connected in the series field circuit for this purpose.

C. J. FUETTERER.

Thomas, W. Va.

## Sheet Steel Magnetized by Rolling?

Can any reader explain why sheet steel becomes magnetized while passing through the rolls? In the rolling mill, cold water is run over the rolls while the billet is "roughed down" and also during the subsequent passes which reduce the metal to its final thickness. After the sheet is passed through the "flattener" it is cut up, and some of it is magnetized and other parts are not. We have never detected any magnetism in steel that has been rolled without the use of water on the rolls.

I will appreciate any suggestions as to the cause of the magnetization.

A. R. COFFMAN.

Scottdale, Penn.

A new method of coating various substances with metals, the invention of a Swiss engineer, consists in reducing molten tin, zinc, copper, lead, aluminum, or other metal or alloy to a state of pulverization by pressure of an inert gas —nitrogen or hydrogen—and in that state driving it against the surface to be covered from a flexible tube with a tip like that of a large vaporizer for handling liquids.

# Gas Power Department

*Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men*

## Some Instructive Indicator Diagrams

By J. C. Parmely

The accompanying diagrams were taken from a horizontal single-cylinder single-acting gas engine working on the four-stroke cycle and show very plainly the effect of changing the point of ignition on light loads. The cylinder is 18¾ inches bore and the stroke is 27 inches. The engine is rated at 100 horsepower and runs at 190 revolutions per minute at full load. The compression is 180 pounds per square inch. The engine is governed by throttling the mixture entering the cylinder, which is accomplished by a butterfly valve in the passage between the mixing valve and cylinder, controlled directly by governor.

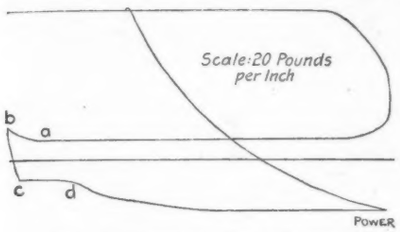



FIG. 1. NORMAL STOP DIAGRAM

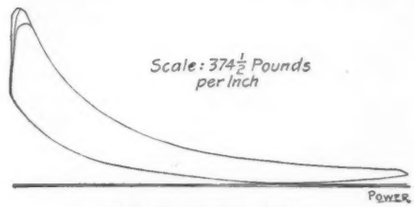



FIG. 2. HEAVY LOAD

The first three diagrams show the operation of the engine under normal conditions. Fig. 1 is a stop diagram taken with a 10-pound spring and it shows that the valves are adjusted properly, with the possible exception of the exhaust valve, which appears to close a trifle early. The rise of pressure from *a* to *b*, however, is very small, being slightly over one pound, and may reasonably be disregarded. The horizontal portion of the first part of the suction stroke, *c* to *d*, is probably due to the inertia of the incoming gases.

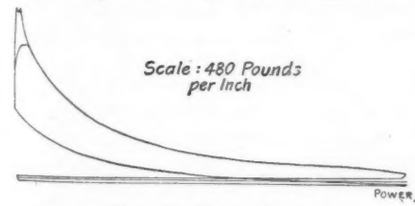



FIG. 3. FULL LOAD

Fig. 2 shows a diagram taken, with a spring having a true scale of 187.2 pounds per inch*, when the engine was carrying a very heavy load. The explosion pressure on this diagram is about 370 pounds per square inch and the pressure at the opening of the exhaust valve is about 23 pounds per square inch and the diagram figures out 136.4 indicated horsepower. This engine frequently carries a load of 80 kilowatts for an hour or more.

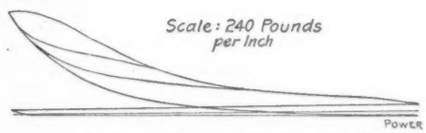



FIG. 4. LIGHT LOAD; SETTINGS UNCHANGED

Fig. 3 shows a diagram taken with a spring having a scale of 240 pounds per inch.* This diagram was taken at a time when the engine was showing very slight signs of preignition, a slight thump being heard. This is indicated on the diagram by the excessive explosion pressure of one of the cycles, which is approximately 450 pounds per square inch.

The next diagram, Fig. 4, shows the effect of a light load. The load at the switchboard was 17.22 kilowatts and the indicated horsepower was 38.7, giving a combined efficiency of 59.6 per cent. for the engine, belt transmission and dynamo. This diagram was taken with a 120-pound spring* and shows a mean effective pressure of 21.6 pounds per square inch. It will be noted that the compression on this diagram is only about 140 pounds per square inch; this was doubtless due to the throttling action of the governor on the light load. The absence of pressure rise after ignition will also be noted. This is probably due to the fact that only a small amount of mixture is admitted by the governor. The action of the governor is also illustrated in this diagram by the variation of the expansion line.

The three remaining diagrams were taken when the load upon the engine was very light, to show the effect of changing the point of ignition upon the operation of the engine. These diagrams were all taken with a 120-pound spring and under conditions of steady load with gas of a fairly good quality. The temperature of the jacket-water discharge was comparatively low at this time.

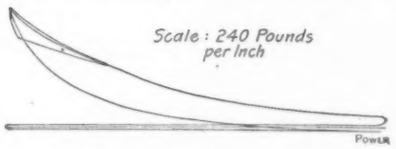



FIG. 5. VERY LIGHT LOAD

Fig. 5 is very similar to Fig. 4 in that there is no rise of pressure due to combustion. The ignition setting was 25 degrees ahead of dead center for both of these diagrams. The small triangular area at the top of Fig. 5 was undoubtedly due to the fact that during one of the cycles shown by this diagram the mixture in the cylinder was poor and did not burn readily. This allowed the pressure to drop immediately after the ignition, causing the lower expansion line to be drawn. On light loads, occurrences similar to this are very common.

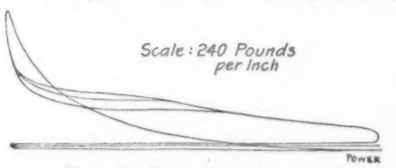



FIG. 6. RETARDED IGNITION

Fig. 6 was taken with the igniter retarded as far as possible from the normal operating position, which was about 25 degrees ahead of the dead center. According to the diagram, the ignition actually occurred slightly beyond the dead center and I think the igniter was set about 10 degrees past the center. The loop in the diagram is peculiar and is probably due to the cooling influence of

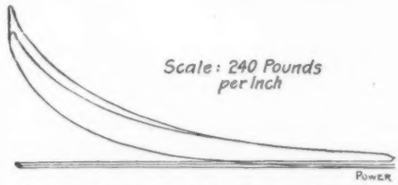



FIG. 7. LIGHT LOAD; EXTRA ADVANCE

*The scales of all of the diagrams as printed here are twice the actual spring scales.—ED.

the jacket water, decreasing the volume of the gas until the moment when ignition occurred.

Immediately after taking this diagram the spark was advanced ahead of the normal operating position to 30 degrees early. The engine did not run steadily at this point but speeded up and slowed down alternately. Fig. 7 shows this effect in the different areas of the two cycles which were recorded.

# LETTERS

## Corrosion of Water Cooled Exhaust Pipe

In Mr. Wild's letter under the above heading in the May 9 number, he does not say whether or not the exhaust pipe

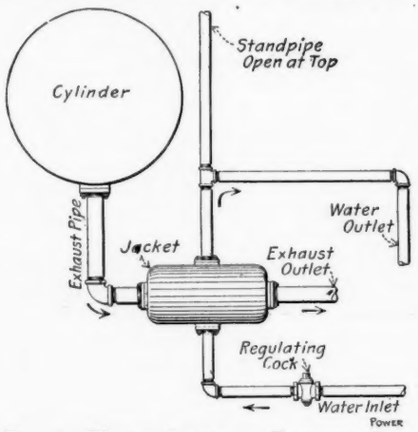



FIG. 1. WATER JACKETED EXHAUST PIPE

and muffler are drained to remove the water of condensation formed when the engine is not running. If the acid liquid cannot be readily discharged it is sure to eat rapidly through the metal. In any case, care should be taken to adjust the water feed to the exhaust pipe so that no excess of water is admitted; only enough should be allowed to pass to insure the water all being converted into steam. If the engine runs on a constant load, this should not be difficult of attainment, but if the load is variable, it will probably be found impracticable to vary the water supply accordingly and in this case the minimum amount of water feed will probably pay best in the long run.

A good way to cool the gases is to water-jacket the exhaust pipe, as shown diagrammatically in Fig. 1, if the water is available. It will probably be found cheaper and more satisfactory to have a short water-jacketed section cast with flanges at each end of it than to have a jacket fitted to the existing pipe, although I have seen the latter method applied with success. In multicylinder exhaust manifolds the piping may be arranged as in Fig. 2, with or without the standpipe, according to necessity.

Another way to cool the gases and secure efficient muffling, usually adopted primarily for the latter purpose, is to provide a series of iron chambers to allow for the continuous expansion of the gases.

JOHN S. LEESE.

Manchester, Eng.

## Mr. Rushmore's Operating Costs

In the issue of May 23, on page 812, appeared "A Comparison of Actual Gas Power and Central Station Figures," by Samuel W. Rushmore. I cannot agree with some of the general deductions reached by Mr. Rushmore, which seem to be more or less in line with the arithmetical errors which he has made. He gives the total cost as quoted by the central station as $555 per month and states that it would be necessary to use $125 worth of gas per month; this makes a total of $680, or 3.4 cents per kilowatt-hour. Mr. Rushmore figures this at 3.9 cents per kilowatt-hour, an error of ½ cent per kilowatt-hour. Mr. Rushmore includes no charges for repairs, cooling water or ash removal, and makes a very general assumption in regard to the fixed charges.

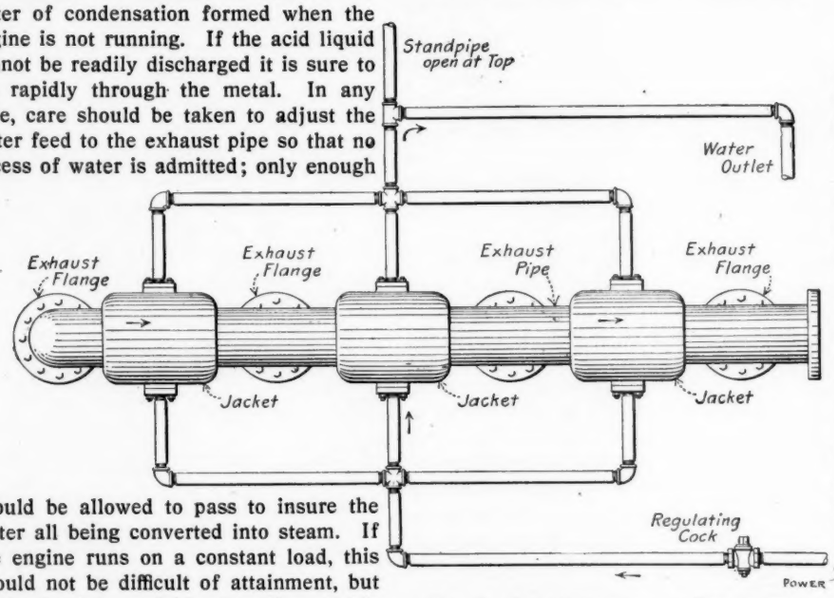



FIG. 2. ARRANGEMENT OF PIPING FOR MULTICYLINDER EXHAUST MANIFOLDS

Apparently the standby losses included are only the night losses during the working days of the week; in other words, the standby costs from Saturday noon to the following Monday morning are not included.

The cost of the plant will be approximately as follows:

| | |
|---|---|
| Producers, 400-horsepower at $12 | $4,800 |
| Engines and auxiliaries, 240-horsepower at $45 | 10,800 |
| Foundations, settings, piping, etc., for producers and engines, 400-horsepower at $12 | 4,800 |
| Building, 400-horsepower at $12 | 4,800 |
| Total | $25,200 |

In a gas-engine plant of this character it is conservative to figure depreciation at 8 per cent., interest at 5 per cent., taxes and insurance at 2 per cent. and profit on investment at 7 per cent., making total fixed charges of 22 per cent., or $5544 per year. It is unnecessary to refer to the items on interest and insurance and taxes, as these are conservative. A plant owner would also hardly consider investing $25,000 in any branch of his business unless he expected to make some profit on the investment and 7 per cent. is a very conservative figure.

From a number of tests on operating plants, three cubic feet of cooling water per kilowatt-hour is a good average figure. During the six-day test referred to by Mr. Rushmore he would probably use 15,280 cubic feet of water, which, at 90 cents per 1000 cubic feet, comes to $14.75.

The corrected operating costs will then be:

| | |
|---|---|
| Coal, 15,218 pounds | $21.50 |
| Cylinder oil | 3.00 |
| Engine oil | 1.25 |
| Kerosene | 0.20 |
| Waste | 0.80 |
| Labor | 33.00 |
| Cooling water | 14.75 |
| Total | $74.50 |

Kilowatt-hours generated, 5094; operating cost per kilowatt-hour, 1.463 cents; fixed charges per week, $106.50; fixed charges per kilowatt-hour, 2.09 cents.

Mr. Rushmore has also figured the cost of central-station power for a rated capacity of 350 horsepower, whereas the power delivered by his gas engine was only 180 horsepower. Neither has he made any allowance for emergency service or breakdown service. A number of instances have come to my notice of gas-engine plants installed within the last few years where the emergency service has been very expensive, in spite of the efforts of capable operating engineers and a large amount of time devoted to the matter by the plant owners. To quote the words of the owner of a 240-horsepower gas-engine installation: "Our electricity did not cost us very much when we had it, but it cost us a whole lot more when we did not have it, and each time the wheels stopped going around the central-station service looked more attractive than ever, so we have sold the plant, and you can see how the factory is running."

If Mr. Rushmore's plant were located in Rochester, we could give him a rate

based on his six-day plant test very much lower than 3.6 cents per kilowatt-hour. I am afraid he is leaving out many of the items entering into the total cost of his power when he makes the statement that it is not costing over one cent per kilowatt-hour. If he were considering his operating charges alone, and making no charges for repairs, or ash removal, or overtime on the part of the plant operators, and crediting the operating cost for the week, as shown above, with $30 for purchased gas, his net cost per week would be $44.50, or 0.874 of a cent per kilowatt-hour. The fixed charges on top of this makes a total of 2.964 cents per kilowatt-hour, instead of "not much over 1 cent per kilowatt-hour" as stated by Mr. Rushmore.

R. D. De Wolf,
Ass't Mech. Engr., Rochester Ry. and Light Company.
Rochester, N. Y.

Mr. De Wolf's criticism of my figure of 0.39 cent per kilowatt-hour is correct; I find I divided the total cost by the previous month's meter record of only 17,500 kilowatt-hours.

Mr. De Wolf is wrong in saying that I included no charge for repairs; I stated that our repairs and adjustments for a period of two years had not exceeded $10 per engine every sixty days.

I made no charge for cooling water because we have our own water supply and the engine consumption is a small part of our total pumpage. The coal handling and ash removal are included in the wages of the producer man; the coal is dumped directly from the railway trestle into the producer charging car, and contractors filling in land call for and remove the ashes free of charge.

I clearly stated that as we would not dismantle the gas-power plant should we adopt central-station service, therefore I did not include the otherwise important items of interest, depreciation or amortization.

I neglected to state that the central-station figures were based on the installation of a 150-horsepower alternating-current motor coupled to a direct-current generator, and were not based on the full use of the 350 horsepower total rating of our motors, although the two engines used in the test totaled 240 brake horsepower. Repairs, depreciation, interest, etc., on a motor-generator would wipe out any item I may have overlooked in present cost.

Although these two engines have been in constant service without any serious shutdown since first installed, one more than four and the other nearly two years ago, we have an additional 75-horsepower reserve gas engine and a 100-horsepower steam plant which we operate during the heating season.

We have no central-station breakdown service, yet, with the two Körting engines alone in service, our machinery has operated during the last three years with no more interruption than suffered by other shops in this district supplied by the central station.

I was a pioneer in the adoption of producer gas more than eight years ago. I have been through all the harrowing experiences and my present plant investment is probably twice as great as would be required today for equal results. Still, if I were now to make a fresh start in this locality, I would unhesitatingly install gas equipment and with the full expectation of producing power at less than half the central-station figures.

Of course, I regret that I am unable to hook on to Mr. De Wolf's busbars at Rochester, for nearly everyone knows that in that favored city central-station power is much cheaper than in most other localities.

Samuel W. Rushmore.
Plainfield, N. J.

## Gas Production from Crude Oil

Reading E. C. Jones' paper presented at the San Francisco meeting of the American Society of Mechanical Engineers and printed in Power for May 2, I was somewhat surprised to note that Mr. Jones went to the expense of compressing air to 35 or 40 pounds pressure for the injection of the oil and to assist in the partial combustion of the oil in his experiments with an oil gas generator. We find it unnecessary, but prefer to use steam, as the energy to atomize the oil with it practically costs us nothing and it adds stability and volume to the generated gas. It also cuts down the nitrogen content away below Mr. Jones' figure of 70 per cent. An economizer in our exhaust gases gives us all the energy required for atomizing, and in this way heat that would otherwise be lost is utilized.

Of course, we get more hydrogen in the gas by using the steam, but I do not consider that a disadvantage. H. F. Smith, of the Smith Gas Power Company, stated in a paper read last year before the National Gas and Gasolene Engine Trades Association that he had never known of a single authentic case of preignition due to the presence of hydrogen; he had personally observed the operation of an engine on which the compression was run up experimentally to as high as 220 pounds per square inch, while supplied with gas containing 27 per cent. of free hydrogen, without any sign of preignition.

Our experience agrees with Mr. Smith's. We are now distributing to several thousand consumers illuminating gas under pressure, by means of pumps driven by gas engines running on crude-oil gas in the production of which we use steam as the atomizing agent and there is no preignition; neither is there any deposition on the cylinder wall or inlet valves. The spark plug has been in use two months at a time and on being examined has been returned for further service.

Joseph J. Nix.
Los Angeles, Cal.

## Recovering Used Oil

Draining the oil channels of gas-engine beds and other means of keeping oil from flying in all directions seem to be ignored by most builders, more especially in the smaller types of engines. Though there have been feeble attempts in this direction, the results are not very satisfactory. The difficulty is increased in the vertical multi-cylinder engines using the splash system of oiling. The editorial paragraph in a recent issue of Power hits the nail on the head exactly in saying that splash lubrication is in the same class as hot-tube ignition.

I have handled several makes of gas engines and have failed to find one that could be kept clean of surplus oil any length of time. The only economical method is to use the oil over again by collecting the drips and circulating the oil through a filter. For example, if the excess oil of a single-cylinder horizontal engine were collected and filtered, the surplus from the new oil fed to the piston would provide a continuous supply for the bearings.

If gas-engine operators would figure up their oil bills, then collect all of the surplus oil and filter it, they would find that the filter had paid for itself by the time the first barrel was used.

There are various ways in which oil can go to waste. Usually the piston and the crank disks are not covered properly and the draft created by the motion of the piston will throw the oil on the flywheels. In a great many cases there is no provision for draining the channels around the base of the engine and it is a difficult task to drill and tap it for pipes if the channel is divided up by foundation-bolt bosses. In such a case, a neat job can be made with a sheet-iron trough extended along the edge of the channel supported by clips made of sheet iron. The clips should flare up at the edges and three or four strands of candlewick can be laid on the clips so that oil will drain from the channel by capillary attraction.

If the oil has a tendency to work out of the bearing along the shaft, a good plan is to make a spring wiper and fasten it to the engine beneath the shaft so it will drain into the channel.

While these simple suggestions are crude, they are effective in practice.

N. E. Woolman.
Danbury, Ia.

# Readers with Something to Say

*Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words wanted*

## This Engineer Made Good

While working in a colliery as a ventilating-fan engineer, a young man was sent to the power house to take the place of one of the engineers. He was not skilled in running electrical machinery or steam turbines, but started in to do his best.

There were two 100-kilowatt turbine sets, the bearings of which were being melted out at the rate of three or four a week and costing the company $15 for each bearing. Vanes were also broken and the shafts were sprung.

He first cleaned the dirty oil cups that oiled the bearings. As a result, during the first month only three bearings seized, but not seriously, and they were scraped and put back. He refused to take heavy oil the storekeeper gave out and by using a lighter grade reduced his oil consumption from 12 to 7½ gallons a week.

One of the centrifugal condenser pumps refused to pick up after being shut down. It was packed and repacked by half a dozen different men, but was not improved.

The young engineer, having concluded that a porous casting caused the trouble, gave the casting one coat of quick-drying iron oxide and two coats of shellac.

When the engine-room force came on at 6 a.m. the pump was working well and the trouble ceased.

A few weeks later the lights went out from loss of vacuum and low steam pressure. The regular engineer struggled along with a 15-inch vacuum and the voltage was reduced from 500 to 420. It was believed that the plant would not work under 120 pounds steam pressure.

When the regular engineer went off duty the young man noticed that the feed pump was running overfast, and that the condenser pump was running too slow. He speeded it up and got a 27-inch vacuum. As the pump had a peculiar sound he concluded that it was not getting enough water. A boy was sent to the canal to clean the strainer which was found to be covered with a mass of weeds. The result was that the feed pump was slowed down, the voltage was back to normal and the machines carried their loads with ease.

This experience goes to prove that it pays to always be on the alert and find out the why of things about the plant.

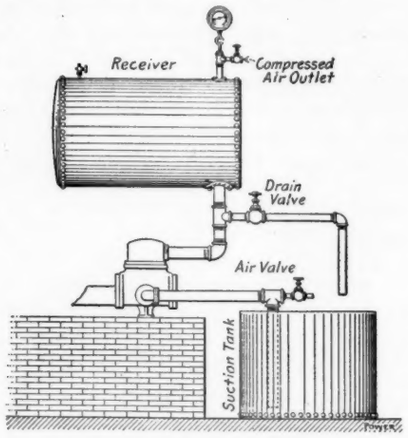
J. P. HUGHES.

Toledo, O.

## Pump Used Compressed Air

Some time ago I was employed in a plant where iron barrels were occasionally tested, an air pressure of 10 or 12 pounds per square inch being necessary. The foreman said he had thought of using a small duplex pump, but he could not pump over 4 pounds of air pressure. I told him I could get all the pressure he needed and, to prove my argument, devised the scheme shown in the accompanying drawing.

The barrel to be tested is first connected by a hose to the open-air outlet




PIPING FROM PUMP TO TANKS

valve on the receiver. The drain valve is closed. The pump is then slowly started and when primed the air valve on the suction line is opened just enough to prevent the pump from entirely "losing its water." By proper regulation of this air valve the pump will take in a large volume of air with each stroke and just enough water to keep the plungers and valves fairly well sealed. When a pressure of 8 or 10 pounds is reached the air valve on the suction line is closed, the pump takes water and the receiver is nearly filled. This forces the air out of the receiver into the barrel being tested and increases the pressure at the same time.

Should more pressure be desired the air-outlet valve is closed and the receiver is drained into the suction tank. The small valve shown on top of the receiver admits air when the receiver is being drained. The operation mentioned is then repeated.

Incidentally it is not the most economical way of compressing air.

LOUIS T. WATRY.

Pueblo, Colo.

## Filing Engineering Articles

One of the inconveniences in filing magazine clippings is that of having two or more articles, on different subjects and desired for filing, printed on one sheet. This precludes the collecting of all articles on one subject under a single head. The easiest plan would be to purchase as many copies as there are articles desired; but, unfortunately, few men can afford this.

The envelop system has several disadvantages: It is bulky and much time is expended in getting a clipping out of its envelop, which if used frequently is soon destroyed.

Clippings laid flat between covers and held in place as in a book are preserved with the least wear and tear. A very convenient file for this purpose is that commonly known as a loose-leaf grip file. This consists of a piece of tough manila paper, folded to form a cover. To this is securely fixed a flexible metal cup with two needles, over which the papers to be filed are placed after being perforated with a punch supplied for the purpose. The needles are then bent down and outward, the locks secured, and the filing is complete. The files, which take up only the room of the papers themselves, have expanding backs to accommodate the growth of the contents; each file holds about 300 papers.

These files are made in several sizes, but I find the 12x9-inch size the most convenient.

It is not always advisable to cut away the reading matter surrounding an article; some of these pages should be left intact and the reading matter not required crossed out. These marked pages may then be used as a background upon which to paste small clippings.

The files should have a complete and reliable index, the card index being the most suitable for this purpose. This requires blank cards on which to record the articles, a set of alphabetical guide cards, for locating the index card on

which any article has been recorded, and a box in which to keep the cards.

To illustrate the use of the system, assume that it is being started with one file (marked *A*), and it is desired to file the article entitled "Condensers for Steam Engines and Turbines," by Frank Foster, printed in POWER for December, 1906. Having cut it out of the magazine, punched and filed it, and numbered the pages 1, 2, 3, 4, etc., the next step is to index the clipping. This will require four cards as follows:

ARTICLE: Condensers for Steam Engines and Turbines.
AUTHOR: Frank Foster, POWER, December, 1906.
FILE: A. PAGES: 1-4.

ARTICLE: Engines and Turbines, Condensers for Steam.
AUTHOR: Frank Foster, POWER, December, 1906.
FILE: A. PAGES: 1-4.

ARTICLE: Steam Engines and Turbines, Condensers for.
AUTHOR: Frank Foster, POWER, December, 1906.
FILE: A. PAGES: 1-4.

ARTICLE: Turbines, Condensers for Steam Engines and.
AUTHOR: Frank Foster, POWER, December, 1906.
FILE: A. PAGES: 1-4.

The titles ARTICLE, AUTHOR, FILE and PAGES on the cards should be made easily distinguishable from the rest of the record, such as underlining, the use of a different colored ink or by the use of a rubber stamp. All that remains then is locating these four cards in their alphabetical order under their respective guide cards, *C, E, S* and *T*, in the index case.

The initial cost of such a system is 78 cents.

E. A. ANDREWS, JR.

Youngstown, O.

## Sump System for Oil Seepage

For some years the waters in northern California, principally the Sacramento river and its tributaries, have been polluted by seepage of fuel oil from the many industrial plants in the valley.

The Southern Pacific Railroad has been an especially large loser, and, with the view of operation economy, has, after repeated experiments, apparently solved the problem by the installation of a simple sump system at the shops.

Along the bank of the river, opposite the shops, a trench was dug for about 450 feet. In excavating, various strata of oil were exhibited, an accumulation of many years' waste, which percolated through the soil and emerged into the stream from the bedrock. The company constructed a system of seven sumps to engage the oil which, whenever conditions warrant, is skimmed from the surface of the water. For temporary experiment these sumps have been constructed of sacks of sand, but the method has proved so successful that concrete tanks will be built to act as a positive catch basin for all seepage.

To give an idea of the enormous loss of oil from this plant the company within a single week reclaimed fully 350 gallons by its skimming process. At the reduced cost of petroleum to the railroad, this amount is almost sufficient to pay for any cost of installation.

W. A. LAWRENCE.

Los Angeles, Cal.

## Thin Fires

My method of obtaining economical results in the boiler room is to carry the furnace fire just thick enough to cover the grates, and to fire light and often. This gives the coal a chance to burn. I admit enough air through the fire-door baffle plates to get proper combustion.

The firemen keep the ashpit doors so adjusted that there will be an opening into the ashpit just one-fifth of the area of the flue space for the admission of air. I find this to be a very good method of regulating the air, and it saves fuel.

W. T. HURD.

Bellefontaine, O.

## Clogged Condenser Gage Pipe

Recently I had charge of a cross-compound engine and a jet condenser pump. One day as I stood looking at the vacuum gage the pointer dropped to zero and rested on the pin. The pump was working normally and the engine kept on running as if nothing had happened.

Then I said to myself: "The pump is working all right, and there is a good vacuum in the engine cylinder or the engine would let me know it very quickly. The trouble must be between the condenser and the gage."

I took down the ¼-inch gage pipe, which was about 3 feet long, and found that it was full of mud and slime. I cleaned it and put it back; as a result I got 26 inches of vacuum on the gage.

The condensing water was taken from the north branch of the Hoosac river, which after a rain is dirty. Shortly after, I went to a larger plant. Three months later my old superintendent asked me to go down to the mill with him and look over the condenser as he could get no vacuum and the mill had been shut down. I went to the plant and found the engineer and two pipers who had inspected the piping from condenser to the engine for leaks. They found everything in good condition at the water end of the condenser pump but there was no vacuum.

I told the engineer that the superintendent had asked me to come in and help out if I could and that sometimes two heads were better than one.

We started the pump and it ran as nicely as any pump could and the quick start and slow ending of the stroke showed that it was producing a good vacuum in the condenser. The pointer on the vacuum gage never moved, however. I told the engineer that he would find the trouble in the ¼-inch pipe between the gage and condenser.

When he took the pipe down he found it full of slime and mud. After it was cleaned out and replaced he had 23 inches of vacuum showing on the gage with the engines standing still.

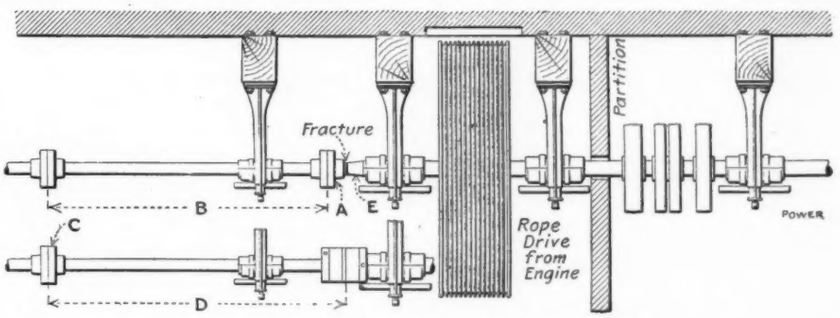
The trouble was found and overcome in ten minutes by one who had had experience in that line. This incident goes to prove that to retain a competent engineer he must be paid what he is worth to his employer and outside assistance would not then be necessary.

JAMES MITCHELL.

North Adams, Mass.

## Line Shaft Repair

An accident occurred in the plant in which I am employed when the main driving shaft carrying a weave shed




BREAK IN THE LINE SHAFT AND THE REPAIRS

twisted off almost flush with the flange coupling *A*, as shown. This shaft was 4½ inches in diameter, but had been turned down to 3 inches where it fitted in the coupling. To take this shaft out would have meant shutting down the entire weaving department for the rest of the week, as the rope drive would have

had to come off all the ropes unspliced, as well as taking off the driving pulleys for the other room.

The repair job was begun by taking out the section of shaft *B*, on which were two flange couplings and to which the coupling on the broken piece was bolted.

The flanges were removed from this shaft, and the flange *C* was then pressed on a longer section of shaft *D*, which came up to the point where the shaft had twisted off.

A compound rest was then taken from a lathe and rigged up on a staging alongside the shaft and that part of the tapered shaft at *E* was turned down to a diameter of 3 inches.

This turning was done during working hours, with the other room running at full speed, the shaft making 270 revolutions per minute.

A keyseat was cut at night in the broken shaft with a portable keyseating machine, and the two shafts were then connected by a box coupling, as shown in the lower view.

ROBERT A. BOND.

New Bedford, Mass.

## Corliss Valve Setting

Methods of setting the valves of a Corliss engine have often been described, but nothing has been said that would be

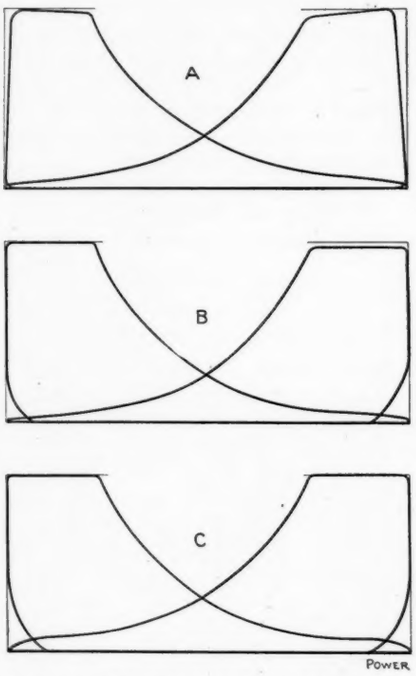



FIG. 1. DIAGRAMS TAKEN FROM THE ENGINE

of service to the average engineer if, after having set the valves according to laid-down rules, factory marks or blueprints furnished by the manufacturer, the engine would not work satisfactorily. In my experience factory marks and blueprints have often been disregarded and the valves set to suit the indicator as well as to obtain smooth running.

One engine, of the heavy-duty type, 38x40-inch cylinder, was direct connected to two double-acting 20x40-inch ammonia compressors, working against 160 to 180 pounds discharge pressure. The indicator showed an initial pressure of 110 pounds per square inch. The steam valves were of the slotted type, that is, live and exhaust steam traveled through the valve instead of over it. The directions for setting these valves were as follows:

Set the steam valve with ½-inch lap, exhaust valve ⅛ inch lap and give the steam valve $\frac{1}{16}$ inch lead. The blueprint showed the amount of travel for each valve and length of rods from the wrist-plate to the bell crank. After the valves were all set according to directions and the eccentric fastened, the engine was turned by hand to the opposite center and the lead adjusted to suit the $\frac{1}{16}$-inch mark which, of course, altered the lap slightly. The engine was started up, and if one ever heard a bad running machine this one was it. An indicator was attached and a diagram produced, as shown in diagram *A*.

I stopped the engine and examined the diagram. The thump occurred after the piston had traveled 2 inches from either end, and I wondered why the thump was not at the beginning of the stroke. If it was due to lack of compression, the thump would have happened before it reached the end of the stroke; if it were due to too much lead, it would have happened at the beginning of the stroke; but coming when the piston was 2 inches from either end, when the steam valve was nearly open, I will admit I was puzzled.

I reasoned that the thump was due to concussion by having full steam pressure admitted to the cylinder too late, as the indicator showed. I advanced the eccentric slightly and the thump lessened, and advancing the eccentric a little more the thump ceased, but the exhaust valves rattled slightly. I took another indicator diagram and *B* was the result.

I had followed the marks on the valves and cylinder, also the blueprints and instructions of the builder and I had no compression, even after altering, so to get the shaft, bearings, crank and crosshead running quietly, I started to lengthen the reach rods on the exhaust valves until both valves had ⅜-inch lap. Then, to get both ends alike, I lengthened one rod until one exhaust valve had $\frac{7}{16}$ inch lap, obtaining diagram *C*.

This adjustment produced a smooth-running engine, but the speed could not be changed more than 10 revolutions per minute, and in this plant the ice machine should run at 25 as well as at 70 revolutions per minute. The regulating arm on which the weight hung for changing the speed was changed from its original shape to look like that shown in Fig. 2. Then a weight of equal size was put on each side. This arrangement worked all right at speeds from 38 to 50 revolutions per minute. I next attached a spring, as shown. This spring attachment solved the problem and the engine

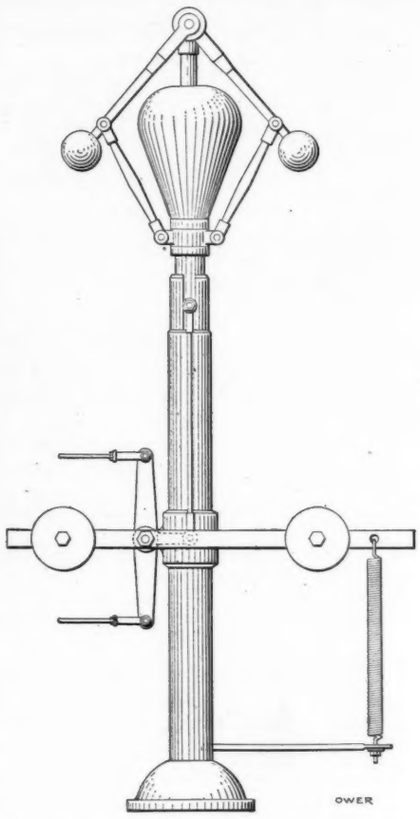



FIG. 2. SPRING ATTACHED TO THE GOVERNOR ARM

can be run from 10 to 50 revolutions per minute, with the throttle wide open and with a steady cutoff.

W. NOEYES.

Kansas City, Mo.

## Reinforced Crank Pin Brasses

The crank-pin box on a small steam engine gave trouble by heating. On examination it was found that the brasses were so lightly made that under heavy loads they would spring. There was a center rib on each against which the rod end and adjusting wedge bore, and metal on each side of this rib was left very thin. As a consequence this rib was forced against the crank pin, causing the center of the brasses to wear faster than the outer ends.

The cavity on each side of the rib was cleaned, turned and then filled with hard babbitt metal. This gave a full bearing for the rod end and adjusting wedge. After making this change the box gave no further trouble.

JAMES W. LITTLE.

Fruitland, Wash.

# Questions Before the House

*Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues*

## Size of Turbine Exhaust Pipes

William Kent says that my formula, given in POWER of March 28, appears to be defective insofar as it does not include the length of the pipe and the allowable drop in pressure between the condenser and the turbine.

As to the length, I will be pleased to consider suggestions for making the area of the pipe vary with variation in the length if I am shown that this is necessary or desirable. My present opinion is that the length affects the desirable area very little if at all.

Consider an exhaust pipe 10 feet long and of any reasonable sectional area. This pipe costs a certain sum of money, and the space it occupies may cost something. If we were to reduce the sectional area of the pipe, we would reduce the cost.

The friction of the pipe causes the vacuum at the exhaust end of the turbine to be less than the vacuum at the condenser and the difference in the vacuum will depend on the area of the pipe. The pressure required at the turbine is, say, $P$. To obtain this pressure, a lower pressure, say, $P_1$, must be maintained at the condenser; and the difference in pressure, or $P - P_1$, depends on the area of the exhaust pipe.

If we were to reduce the area, we would increase $P - P_1$; and, therefore, to maintain $P$ constant, we should require to reduce $P_1$ and this would involve increased initial cost of condensing plant, or increased running charges, or both. What is required is to determine the diameter of pipe at which the rate of reduction in cost of pipe with reduction in area will just equal the rate of increase in cost of condensing plant with reduction in area.

We have been considering a pipe 10 feet long but, if one only 5 feet long is taken, the variation in cost of pipe with reduction in area would be just half, while the increase in cost of the condensing plant would also just be half, so that the best area for the 5-foot pipe would be exactly the same as the best area for the 10-foot pipe. It therefore appears that the length of the pipe does not require to be taken into account.

As regards Mr. Kent's other objection to my formula, namely, that it does not include the allowable drop in pressure, it would, I consider, be foolish for anyone to state an allowable drop of pressure without considering the cost of obtaining this. The formula is intended to determine the area which will give the drop of pressure which is best in any case. The drop will, of course, be very much greater for a long pipe than a short one. For a short pipe the drop ought to be very small because it can be kept small at a trifling cost. For a pipe twenty times as long, it would be unreasonable to limit the drop to the same amount, as this could only be accomplished by multiplying the sectional area of the pipe many times and the cost would be enormously increased.

As regards the exponent 0.4 of the factor $f$ in my formula, most formulas for the flow of fluids through pipes of circular section make the friction vary as the square of the volume of fluid passing a given point in the pipe every minute, and also make the friction vary as the fifth power of the diameter of the pipe. With nonelastic fluids weight may be, and often is, substituted for volume in the formula. If $F$ denotes the friction, $B$ the volume and $d$ the diameter

$$F \propto \frac{B^2}{d^5}$$

For convenience later on, let $A$ denote the internal sectional area of the pipe, then

$$F \propto \frac{B^2}{A^{2.5}}$$

If $v$ denotes the velocity of the fluid

$$B \propto vA$$

Therefore

$$F \propto \frac{v^2}{A^{0.5}}$$

That is, for constant velocity the friction varies inversely as the square root of the sectional area. Presumably the reason is that the ratio of superficial area of the pipe per foot length to sectional area varies inversely as the square root of the latter, and the friction increases with the superficial area. Therefore, for pipes of dissimilar but equal sections, we can take the friction to vary as the superficial area per foot length, or simply as the internal periphery.

If $f$ denotes the periphery in feet of a figure of the shape of the cross-section of the pipe or duct and of an area of one square foot, then for all pipes of one square foot sectional area

$$F \propto f$$

and for all pipes whatever

$$F \propto \frac{B^2 f}{A^{2.5}}$$

Therefore, for equal friction

$$A^{2.5} \propto B^2 f$$

or

$$A \propto B^{0.8} f^{0.4}$$

In my formula $W$ (the total weight of steam per hour) is employed for convenience instead of $B$.

R. M. NEILSON.

Glasgow, Scotland.

## Did Not Hook On

I submit the following answers to the questions of S. E. Mead as given in the May 23 number, page 814, regarding the diagram from a Corliss engine which did not hook up.

"Why does the diagram from that end include any area?"

Because the movement of the piston changes the pressure of the contents of that end of the cylinder by alternately compressing and expanding it.

"Why does not the expansion line follow back upon the same line as the compression?"

The expansion line drops down below the compression line because at the beginning of this stroke both the steam and the exhaust valve at this end of the cylinder are closed, and as the piston starts away the contents of the clearance space expand, thus losing pressure very rapidly at first, and then gradually until at about half stroke, after which point it runs nearly parallel with the line of zero pressure. Then when the exhaust valve opens it raises slightly to meet the line of lowest pressure attained by the condenser.

In explanation of this diagram I would say that it is directly opposite to the ordinary steam-engine diagram in that its expansion curve is the lower one which approaches nearest the line of absolute vacuum, and the compression curve is the upper one or the one nearest the atmospheric line. In this the diagram is very similar to the diagram of an air compressor.

The compression starts in a line very nearly straight and parallel to the atmospheric line for more than half stroke

This is because the exhaust valve is open and the piston moves against the lowest vacuum attained. Then as the exhaust valve closes, this charge of low-pressure steam is compressed, thus raising the pressure highest at the end of the stroke.

In answer to Mr. Mead's last question, I would say that the expansion line cannot be lowered as it represents the very best possible vacuum attainable with his engine.

CHARLES F. CLARK.

Hartwick, N. Y.

## Boilers Foam

In the May 23 number, J. M. Stewart asks for opinions about his foaming boilers. I have had the same trouble with domeless boilers connected to hoisting engines, and have noticed that sometimes they do not furnish dry steam even on more regular service.

I believe that something serving the purpose of a dome is necessary on a horizontal tubular boiler, but I would not want a dome connected in the usual way, as it greatly weakens the shell to cut out the area necessary for the rivet holes and steam passage.

I would prefer a horizontal drum connected to one end to the boiler by a thimble and flange joint, and at the other end resting freely on a support made by riveting thimbles of the proper length to the shell of the boiler and drum.

A drum connected in this way will cause some loss from radiation but if it is properly covered the loss will be small and the extra steam space is a great gain to a boiler connected to a hoisting engine.

Mr. Stewart's trouble might be in his feed water, but if he will boil samples of it in an open vessel, he will be able to tell by its behavior if such is the fact.

H. L. TURNER.

Bartlesville, Okla.

## Receiver Pressure

L. Johnson, in the May 16 issue, asserts that the most economical receiver pressure is that which causes the governor to revolve in the highest plane, basing his conclusion on the fallacy that the earlier the cutoff in the high-pressure cylinder the smaller the steam consumption. The fact is, early cutoff in the high-pressure cylinder does not necessarily mean less steam consumption, and furthermore it is possible with no change of cutoff to vary the steam consumption within considerable limits, say 10 per cent., simply by changing the receiver pressure.

The steam in the cylinder at cutoff is made up of two parts, a small amount left in the cylinder at the closing of the exhaust valve, and a larger amount which flows in from the chest while the admission valve is open. These two volumes of steam mingle in the cylinder, but they can be kept separate for purposes of reasoning. The steam left in the cylinder at compression is sometimes called cushion steam. Its volume is constant but its weight is variable depending on the density, which in turn varies with the receiver pressure. With high receiver pressure the cushion steam may be 15 per cent. of the steam at cutoff, while with low receiver pressure it may be as low as 5 per cent. These figures are based on 5 per cent. clearance and one-quarter cutoff and are to be regarded as approximate values for average conditions.

The steam consumption is measured by the steam that comes in from the chest and joins with the cushion steam to make the volume present at cutoff. Calling the weight of steam at cutoff 100 and the cushion steam 5, the balance, or steam consumption, is 95 corresponding to low receiver pressure. Calling the steam at cutoff 100 and the cushion steam 15, the steam consumption is 85 for a high receiver pressure. In this case the steam consumption is changed 10 per cent. by variation in the cushion steam, brought about by alterations in the receiver pressure and not by a change of cutoff in the high-pressure cylinder.

Experience tends to confirm the truth of the foregoing reasoning. A good illustration is to be found in some forms of pumping engines having no governor, where the high-pressure cutoff can be set by hand. Under these conditions with a steady load, if the receiver pressure is lowered the engine gains in speed, showing that it is using more steam and doing more work. If the receiver pressure is raised the engine loses speed, showing that it is using less steam and doing less work.

The simple rule quoted by Mr. Johnson is sure to lead to too low a receiver pressure and hence to poor economy. There is reason to fear that many engineers are following it, not knowing the unsound basis on which it rests. A final argument against the rule may have weight with those who do not follow the theoretical reasoning. It is that, in all trials of compound engines where the steam is measured and the best performance is desired, the highest economy is obtained with a receiver pressure giving but little drop in pressure at the end of the expansion in the high-pressure cylinder.

E. H. LOCKWOOD.

New Haven, Conn.

## Who Is Responsible?

After a man has worked for you for six months or a year and shown you that he possesses the ordinary amount of brains, then when the water column becomes clogged, instead of getting word to you, unscrews the plug in the column tee and rams a rod through into the boiler, cleaning out the obstruction and also—well, you can imagine the rest; who is responsible? I know men to whom you can talk and explain till you are blue in the face and they will say "yes, yes, I know," and yet by only the fraction of a minute did I save one of them from the disastrous results of taking the top from a check valve before he had closed the globe valve in front of it. However, is there a man in the business who cannot point to some fool thing he has done at some time or other.

C. A. SCOTT.

Wales, Wis.

## Relative Size of Compound and Simple Engine Cylinders

In POWER for April 25, C. E. R. asks, "What would be the comparative diameter of the low-pressure cylinder of a compound engine to develop the same horsepower as a simple engine at the same speed and steam pressure?"

The answer to the question begins with the remark: "If the work done is to be the same in both cases, the number of expansions must be the same." This is equivalent to saying that all steam engines of the same power, speed and steam pressure have the same number of expansions, which, of course, is erroneous.

The reply then continues: "Consequently, with the same initial and terminal pressures, the diameter of the low-pressure cylinder of the compound must be equal to the diameter of the single cylinder of the simple engine." This is approximately true, barring the terminal drop in the high-pressure cylinder and other losses. The question must have been misunderstood. One of the main reasons for compounding is to increase the number of expansions without shortening the cutoff beyond the economical limit, and the terminal pressure in the low-pressure cylinder of a compound may be, and practically always is, considerably lower than that of a simple engine working between the same pressure limits.

As an example, assume a simple noncondensing engine of 500 indicated horsepower, with 140 pounds absolute initial pressure, 16 pounds absolute back pressure; a clearance of 5 per cent. and compression at nine-tenths of the stroke. Cutoff at one-fourth stroke is common for the rated load of such an engine. Then with a diagram factor of 0.85 and a piston speed of 750 feet per minute, the cylinder is 22 inches in diameter. A compound, noncondensing engine of the same power and piston speed, with the same pressure limits, clearance, compression and diagram factor, but with

a terminal pressure of 20 pounds absolute, would have for the diameter of the high- and low-pressure cylinders respectively, 17 and 29 inches. Or, using Tribes tables with a still lower terminal pressure, the cylinders would be 18 and 30 inches in diameter.

Twin engines are sometimes converted into compounds by replacing one of the cylinders by a low-pressure cylinder, the size of which depends on whether a condenser is used; but in all cases it is larger than the cylinder it replaces.

WILLIAM E. NINDE.

Syracuse, N. Y.

## Water in Oil

R. C. Monteagle recently made a statement about water in fuel oil putting out a fire and overflowing from the furnace onto the fire-room floor. In externally fired boilers there is danger of such an accident only when starting the fire and the brick is cold. It is then necessary to watch the burner closely, as the flame may go out and allow oil to flow into the furnace. If the torch is applied there will be, of course, an explosion, as the combustion chamber and tubes will be filled with oil vapor. After the oil has been burning for some time, water may put the fire out, but the oil will instantly ignite, as the brickwork is at a white heat. There is usually some water in fuel oil, but not enough to interfere with the operation of the fire.

Norborne, Mo. ANDREW BLAIR, JR.

## Cooling Hot Bearings

In regard to the cooling of hot bearings, I would like to submit my cure which has never failed and which works quicker than anything I have tried.

Mix half and half by volume No. 6 Keystone grease and ammonia and feed through the oiler as fast as possible by drops. If No. 6 grease is not available, common engine oil will do, but it will have to be stirred almost continuously.

THEODORE L. DARLING.

Pascoag, R. I.

## Antifriction Metals

In the May 16 number of POWER, Charles H. Taylor has a letter on the so called antifriction metals which was of unusual interest to me, especially the part in which he states that the general composition for a nonfriction metal should be 76 parts lead to 24 parts antimony. I cannot understand how Mr. Taylor is going to make this metal stand up under all conditions. It is not to be expected that a bearing running at a fair speed, and with little or no shock, will need the same quality of babbitt as a bearing which is operating under severe conditions. I believe if Mr. Taylor would modify his composition to the proportions of not over 20 parts antimony to 80 parts lead he would have a bearing metal that for light high-speed service will be found to give very satisfactory results.

A composition that has given good service under trying conditions is one composed of 5 per cent. copper, 8 per cent. antimony and 87 per cent. tin. Ordinarily this composition is almost prohibitive on account of the cost, and if allowed to run loose for any length of time it will certainly fracture. If it is kept tight on the journal at all times, that is, just free enough to run, it will be found to be all right.

Another composition that will give very satisfactory results when the shock is not too great consists of 20 per cent. tin, 15 per cent. zinc and 65 per cent. lead. It will improve the wearing quality if the metal is covered with a coat of fine charcoal during the time it is in the melting state.

I. B. GREEN.

New York City.

## The Institute of Operating Engineers

There has been considerable criticism of the Institute of Operating Engineers, but if I fully understand the working of the organization it is certainly worthy and along the right lines. No one has anything to say against the homemade engineer. He is usually a good kind, and this association will only help him to be better. I see no need in his feeling that anyone is working a scheme to manufacture engineers to take his place. The homemade engineer who will be alert and look out for his duties and the welfare of his plant need not fear such an association. In fact, he might easily better himself by joining the order.

The association presents an opportunity for progress to those who are willing to study and are reaching upward for the rung on the top of the ladder. Pushing ahead is no disgrace, and those who are contented might well allow progressive engineers this privilege. The Institute of Operating Engineers is the center of a good movement and if some engineers do not appreciate it and fail to join, there is no occasion to retard the movement.

C. R. MCGAHEY.

Baltimore, Md.

## Vacuum Cleaner

S. G. Rose, in the May 2 number, asks how to remove the dust from floors, ceilings, etc., without sweeping. He practically answers his question when he makes inquiry about a vacuum cleaner. I would advise him not to attempt making a homemade affair, when vacuum-cleaning outfits at small cost can be bought ready for use.

It is a well known fact that dust causes much damage to generators and other appliances and it is a common sight to see an attendant around working at generators and motors trying to dislodge the dust with a bellows or a stream of air from a compressor. A brush fitted to a long handle which is hollow and this handle fitted in turn to the vacuum hose will soon remove all dust from the high spots. A regular vacuum broom attached to the hose will be all right for the floor. The principal items necessary are a dust-settling chamber, a brush or cleaner by which the dust is collected and directed into the hose and an exhaust blower.

JAMES E. NOBLF

Toronto, Can.

## Scale in Feed Pipe

In a recent issue an engineer stated that he had a great deal of trouble with scale in the feed pipe of his boilers. In two different plants with which I have been connected we had the same trouble with the internal feed pipes of Heine and of Babcock & Wilcox boilers. The water was taken from the same river and contained a great deal of scale-forming matter which was mostly carbonates and sulphates of calcium.

In one plant the external feed pipes were made of brass and gave no trouble, but the feed pipes inside the boiler were of iron and became plugged up solid in a very short time. The iron pipes were finally replaced by brass pipes of the same size, and there was no further trouble from scale.

In the other plant the external pipes were of iron and after a short length of time became pitted full of holes from $\frac{1}{16}$ to $\frac{1}{8}$ inch in diameter, while the internal pipes of the same material became closed up with scale. As there are two internal pipes in the Babcock & Wilcox boiler we renewed one with iron and the other with brass. In a few weeks the iron pipe was plugged up and the brass pipe remained clear. It was then decided to use brass in all of the boilers, which, as far as I know, never gave any further trouble.

As 4-inch high-pressure brass pipe was considered too expensive for the main feed line, we used extra-heavy steel pipe, and it is doing fairly well.

I have never been able to find out why the scale did not adhere to the brass pipe as it did to the iron pipe, but in this case brass pipe solved a very dangerous as well as troublesome problem. I would suggest that the engineer above referred to try brass pipes on one boiler and note the results, which will give him

an idea as to whether or not this will solve his difficulty.

J. CASE.

Hyattsville, Md.

I am interested in several letters under the above title and think there is quite a little to be learned. I do not agree with Mr. Jahnke as to his two-pipe system, especially the one connected to the blowoff pipe. Combinations usually mean complications. The following system is better than any other which I have seen or heard of:

It is unnecessary to have two separate feed lines to a boiler, but take care of the one. Always enter the feed pipe somewhere on the top of a return-tubular boiler, as it is perfectly accessible, both inside and outside. If hot or warm water is used all the time, it is unnecessary to have a long pipe inside, as it is apt to cause more trouble than it is worth. Run the pipe down to the flues, then use an elbow and a horizontal piece of pipe not over 3 feet long. To avoid depositing the impurities which the heaters and purifiers have failed to collect, on the flues or sheets, fasten a pipe of not less than 3 inches diameter to the flues so that the feed pipe will discharge into it, carrying in with the discharge a quantity of the boiler water. The impurities will collect inside of the large pipe and may be easily removed on cleaning day with a hose or hammer. Let the feed pipe enter the large pipe only a couple of inches; the large pipe should be not less than 4 feet long. If a pipe above 4 inches in diameter is used, place the collector pipe at an angle so that the water will take a spiral course through it.

RAY GILBERT.

Virginville, W. Va.

## Boiler Design

Having read with interest "Desirable Improvements in Boilers," on page 761 of the May 16 number, it would seem that the correctness of the title in every particular might be questioned when viewed in the light of practical experience in boiler operation. Mr. Dean has adopted an old form of longitudinal joint which he believes is best for boiler work. It has overcome one disadvantage in the former style of joint he recommended in reducing the pitch of the rivets along the calking edge, for this must have been a disadvantage in manufacture even when using heavy butt straps.

While it is hard to see any real advantage in Mr. Dean's new selection of joint over the kind of butt-strapped joint in common use in the United States, it would be interesting to know if such a form of joint really shows under test the calculated strength he gives it. The committee of research on riveted joints of the Institute of Mechanical Engineers found that to obtain the full strength of a double-riveted lap joint it was required that the two rows of rivets be sufficiently separated to prevent the joint breaking zigzag, and making sure that the joint would break straight across; the net section of metal measured zigzag should be 30 to 35 per cent. in excess of that measured along the seam. While Mr. Dean's joint omits the figures to determine whether the zigzag distance is what is required or not, it appears to the eye as if he had neglected to consider this fact in designing his joint, and that he had bunched the rivets too nearly together girthwise.

Mr. Dean's attack on the Manning boiler seems entirely unwarranted; it is true that the ogee flange as originally designed was very thin and of such shape that all the movement due to pressure was concentrated along one line around the flange. A few of the flanges of this design cracked in service, but after the thickness was increased and its form changed none have failed; therefore, his remarks on this score do not apply at all to the Manning boiler as now constructed. Anyone of experience can take direct issue with Mr. Dean when he says that the Manning boiler is unquestionably dangerous. It is only dangerous in the sense that any steam generator is dangerous, and it is far safer than the average boiler. Mr. Dean speaks of the behavior of the ogee flange on a destructive test of this type of boiler, and he doubtless refers to the test made at the Bigelow plant last summer. The writer had the pleasure of witnessing this test and he distinctly remembers that the center of the ogee flange rose only about ¼ inch under a pressure of 700 pounds, while at 450 pounds the movement was not measurable. As a check on this the overall length of the boiler did not change until after a pressure of 400 pounds was reached, and it amounted to less than ⅜ inch at 700 pounds.

As boilers of any type are rarely operated for general purposes at pressures exceeding 200 pounds, I think Mr. Dean is drawing entirely on his imagination when he asserts that the explosion at the Amoskeag mills was caused by pressure on the ogee flange. I am ignorant of any facts that will establish his claim, and the long record of the Manning boiler with a minimum of accidents absolutely refutes his statements implying that this boiler has inherent structural weakness as constructed today.

If Mr. Dean was versed in practical boilermaking he would hardly suggest a coned sheet in place of the ogee flange, for any boilermaker can tell him of the difficulties encountered by the introduction of such a shape. While Mr. Dean's intentions are good in advising that many more tubes than customary may be put in horizontal tubular boilers, I think he is considerably exceeding good practice in this direction. It is a well known fact that a large body of water directly over the fire is desirable in this type of boiler, aside from the need of room sufficient to permit ready access below the tubes. With the number of tubes he recommends for the different-sized boilers, the required space would not be available at this point. The water line in his layouts would also be considerably higher than practice has demonstrated to be required for good service. With a 1-inch space between tubes it is difficult to clean a scaled tube sheet, and reducing this distance by 25 per cent. would certainly make it much more difficult, besides weakening the bridging between tubes. I hardly think many engineers will agree with Mr. Dean that his suggestion along this line can be considered an improvement in boiler design.

J. E. TERMAN.

New Haven, Conn.

## How to Condense Steam

In answer to E. G. Eldred's question, "How to condense steam," I think the best plan is to take a barrel that will hold water, fix an overflow to it and let the water enter through a pipe at the bottom of the barrel. Then make a coil of ½-inch or ¾-inch pipe of a few turns that will go in the barrel easily. Put the bottom end of the coil through the side of the barrel with lock nuts and connect the top to the steam pipe.

I have tried this method and considerable steam can be condensed in a short time. If the water is allowed to drip a few inches through the air it will get some air in it and help the taste of it.

E. V. CHAPMAN.

Decatur, Ill.

## Oil Fuel for Steam Boilers

In the May 16 number Mr. Collins says that in burning oil a boiler can be brought up to 150 pounds steam pressure from cold water in less than half an hour if necessary. This may be possible, but it is certainly not the right thing to do. In an externally fired boiler it will take half an hour to get the furnace hot enough for economical combustion. In my experience with burning oil I have found that it takes from two to four hours to get up steam to working pressure from cold water. I do not agree with his statement that 35 per cent. more capacity can be obtained with fuel oil than when burning high-grade coal. This, at least, has not been my experience.

ANDREW BLAIR, JR.

Norborne, Mo.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
122 South Michigan Boulevard, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, December 20, 1910, at the post office at New York, New York, under the Act of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

*CIRCULATION STATEMENT*

*Of this issue,* 31,000 *copies are printed.*

*None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.*

## Contents

PAGE

The Steam Turbine in Germany..... 984
Metal Welding in Germany..... 987
Schedule of Flanged Fittings..... 988
Training Mr. Duffy..... 990
Energy Drop in Steam Turbines..... 991
Pressure Temperature Relations of Saturated Steam..... 992
An Old Time Tide Mill..... 993
Water Jacket Deductions..... 993
Inertia Effects and Shaft Couplings.... 994
Air Cooled Choke Coils..... 996
Factors in Good Service on Transmission Systems..... 996
Interpole Motor Characteristics..... 997
Central Station Service vs. Isolated Plant Operation..... 998
Parallel Operation of Alternators Driven by Waterwheels ..... 998
Starting Large Motor Generators..... 998
Sheet Steel Magnetized by Rolling..... 998
Some Instructive Indicator Diagrams.. 999
Corrosion of Water Cooled Exhaust Pipe 1000
Mr. Rushmore's Operating Costs..... 1000
Gas Production from Crude Oil..... 1001
Recovering Used Oil..... 1001
Practical Letters:
This Engineer Made Good....Pump Used Compressed Air .... Filing Engineering Articles....Sump System for Oil Seepage....Clogged Condenser Gage Pipe....Line Shaft Repair .... Corliss Valve Setting .... Reinforced Crank Pin Brasses .....1002-1004
Discussion Letters:
Size of Turbine Exhaust Pipes ....Did Not Hook On....Boilers Foam....Receiver Pressure....Who Is Responsible....Relative Size of Compound and Simple Engine Cylinders....Water in Oil....Cooling Hot Bearings....Antifriction Metals....The Institute of Operating Engineers....Scale in Feed Pipe ....Boiler Design....How to Condense Steam.....1005-1008
Editorials .....1009-1010
Double-pipe vs. Atmospheric Condensers 1012
Cutting Packing over a Wooden Mandrel ..... 1014
Air in Ice Water..... 1014
Opening an Ammonia Joint..... 1014
Clearance in Ammonia Compressors.... 1014
Temporary Can Repair Kink..... 1014

# POWER

## Accidents and Education

There are two movements in which the engineer is either consciously or unconsciously interested, for they touch his vocation in vital points. These movements are industrial education and the prevention of industrial accidents.

By industrial education is meant that vocational training which gives the worker a knowledge of the principles underlying his trade or calling as well as the definite and complete manual skill which marks the difference between the master workman and the unreasoning routine drudge. The growing importance of the work of the engineer in the industries of production, transportation and distribution is beginning to be recognized by the general public and by those directly interested, and this recognition manifests itself in attention given to license and inspection legislation and in the establishing of schools, classes and lectureships in universities and other institutions for the purposes of giving practical instruction in steam-power plant operation.

It is plain that the engineer is, or should be, interested in this as it tends to put within easier reach than ever before the systematic education needed to advance him in that intimate knowledge of his vocation that makes for success.

This relates directly to the prevention of accidents in an industry that is as wide as the continent. Steam boilers under pressure are no safer in the care of incompetent operators than is dynamite in the hands of children.

Knowledge of the expansive force of steam, of the potential energy of a mass of highly heated water in a boiler and of the inherent weakness of certain forms of construction—with the ability to calculate the strength of materials and joints and the effects of the repeated distortions of the shell of the lap seam—reduces the hazard of this form of construction and discourages its installation.

With every engineer familiar with the weakness of this type of joint and with the courage that comes with knowledge opposing the making and installing of such apparatus and the operation of that already installed, one class of industrial accidents will be materially reduced.

These are subjects which should not be lightly dropped; on the contrary, they should be weighed with careful judgment if the great body of engineers desires to place the vocation of steam-engine operation in the position, in the estimation of the public, to which it is entitled by its importance.

It is a calling that confers dignity on its followers who give to it the best that is in them.

## The New York Edison Company's Advertisement

The question of the isolated plant versus the central station is becoming a burning one the country over. Engines are replaced by motors, engineers by switch-pulling janitors, and every small plant shut down or precluded by central-station service means losses of sales of engines, boilers, pumps, heaters and auxiliaries to commence with, and of oil, packing and all sorts of supplies continuously. There is therefore little sympathy for the central station among operating engineers and manufacturers of material used in isolated plants. The following letter, recently received, is a manifestation of this lack of sympathy:

New York City, June 14th, 1911.
To the Hill Publishing Co.,
New York City.

Gentlemen:—

I, as well as the business people who use your paper for advertising, also the steam engineers of the city of New York, believe that it is harmful and detrimental to publish advertisements of the New York Edison Company, a company that is shamefully taking the bread from many engineers and firemen, as well as our business from us the manufacturers.

If in the next number of POWER you have another advertisement of the New York Edison Company, the matter will be taken up by all the engineers' locals of this city and other States, as well as the manufacturers, asking them not to take POWER, and declaring the same a paper that is against the engineer and manufacturers. Every engineer will be asked to cast out POWER from his engine room as an enemy.

It is the duty of your paper to help the engineers to fight and make resistance against the usurpations of the New York Edison Company, which is a monopoly and detrimental to all. Why not from this date start an article, and help the engineers who are your friends. Write an article, "The Little Engineer vs. The Big Engineer" (the Big Engineer is the New York Edison Company). If you do all of us will help you to show them up.

Yours very truly,
I. A. of E.

We have received numerous other admonitions by mail, telephone and word-of-mouth against the continued publication of this advertisement.

There are two separate, distinct departments to POWER. Into the advertising pages anybody may come at the regular rate, and say anything that is not libelous or indecent and is in the line of power-plant practice. On the other hand, nobody can buy at any price any space

or utterance in the reading columns. What POWER says editorially is based upon the best information obtainable, and inspired by the ambition to offer the best in the way of analysis, suggestion and observation which its editors with their facilities and in the light of this information can produce. It is by these utterances and not by what appears in its advertising columns that the attitude of a paper should be judged. And what has been the editorial attitude of POWER upon the central-station question? Have you read the "Foreword," October 25, 1910; "Central Station versus Factory Plant," February 14, 1911, and March 21; "The Cost of Power," "Interest and Sinking Fund," March 21; "Will an Isolated Plant Pay?", "The Marginal Principle," March 28; "The Central Station versus Isolated Plant," April 18; "The Central Station Could Not Meet His Figures," April 25; "Isolated Power Plant Makes a Good Showing," May 9; "The Cost of Industrial Power," May 30; "Foreword," June 6; "Central Station Failure," June 20, and the "Central Station Viewpoint," June 20. These are only a part of the references which might have been given, but a reading of the above articles will show the stand we have taken on this subject.

We take it that even the members of the I. A. of E. (we regret that we do not know what the initials stand for) recognize that there are places where the central-station service can be used to advantage, and that their organized opposition to the extension of that service applies to situations where power can be produced cheaper than the central station can legitimately supply it. In opposing the aggressions of the central station beyond its legitimate field; in opposing rate discrimination and subterfuge whereby additional load is taken on at less than the cost would be if charged with its proportion of the fixed charges, as are the services of the small consumer; in exposing the sophistries of the solicitor whose gentle job it is to convince a customer that he can make money by buying current for more than he can make it, the engineers and manufacturers of the whole country—for the question is not a local one—will find POWER with them all the time, and if our unknown correspondent has anything which will help in the process we shall be glad to see it.

---

Peat as gathered contains a large percentage of moisture. To dry it by exposure to the sun and air takes a long time, and to dry it artificially takes a good deal of heat. An attempt has been made at Emden, Germany, to utilize this heat by drying the peat in a closed vessel under pressure, the steam driven off being available for power and other purposes.

## Engineering Graduates

Once more the season of college commencements is at hand and hundreds of young men are about to start on an engineering career. The increasing popularity of the engineering courses, as compared with the older professions, gives rise to the remark so often heard: "The field is being overcrowded." Some will contend that such overcrowding is sure to result in fewer opportunities for advancement and lower salaries, while others will argue that there is always room for good men in any field.

Among the large number graduated every year there are undoubtedly many that fail to make good. This, however, is usually attributable to one of two causes: either the individual has not the qualities that make for success, in which case he would have failed in any other line of work, or he has chosen engineering without seriously consulting his natural inclinations. In this connection, it may be said that fully fifty per cent. of the students when entering college do not really know what line of work they want to make their life's vocation. A course is often chosen because it appears popular, its name is attractive, or some friend has had success in that particular line.

A factor responsible for this condition is the age at which the average student enters upon a technical course. If those who intend taking up engineering would spend two or three years, after leaving high school, in shops or construction work before entering college, they would soon find wherein their inclinations lie; they would get more out of their college course, and would be better prepared to attack practical engineering problems after graduation.

It would appear then that some are doomed to failure through causes independent of the supply and demand; but to offset this the demand for technically trained men has greatly increased during the past few years.

Very often the mistake is made of expecting too much of the technical graduate, and much misunderstanding and criticism of the whole educational system result.

## Inefficient Equipment

Economy is the watchword in the power plant where the management and the engineers are wide awake. In others, economy is a meaningless term, and extravagance takes its place.

Most men will have the hole in their pocket sewed up as soon as it is discovered, for fear that a few cents may be lost, but these same men, if owners of a steam plant, will contentedly watch a fireman shovel dollars into a boiler furnace and make no effort to save any of them. A growl will be heard when the amount of the yearly coal bill is ascertained, but beyond "jumping on" the fireman no effort is made to detect and remedy the cause of excessive coal consumption.

Power plants of less than five-hundred horsepower contain engines, pumps, heaters and other power-plant apparatus which are wasteful in the extreme. But the engine turns the wheels, the pump manages to keep water in the boiler and the heater warms the water a trifle above its normal temperature, all of which seems to satisfy those "higher up."

For years the feed water in a certain electric-light plant was sent to the boilers by means of an injector, the water passing through a heater which had become so foul with scale that the water entering the boilers received practically no additional heat above that imparted to it in passing through the injector.

In another small steam plant an old cylinder heater, about ten feet long, lies on the floor alongside the engine. It is inefficient and, although it does take the chill from the water, much of the heat in the exhaust steam that could be utilized if a proper heater were used escapes to the atmosphere.

These two instances illustrate the manner in which the operating costs of a steam plant can run above normal when apparently everything is all right. But if the men who were financially interested in these plants had taken the trouble to look into the matter of feedwater temperatures, as obtained under the conditions found, and compared them with the temperatures obtained in other plants, their eyes would have been opened to the waste of heat and coal.

Thousands of people are ailing more or less—few are physically whole—but they do not know it. When they do, a doctor is consulted in order that the trouble may be removed.

It is the same with steam plants. Few are operating under economical conditions in every particular, and many are "real sick." Their case is not diagnosed, and their ailment is allowed to grow worse day by day.

When power-plant owners and engineers look upon a steam plant as a source of expense, as not producing a finished product, and make up their minds that it can be operated efficiently, the central station will loom less prominently before the engineer's vision and the owner will have the satisfaction of knowing that he can produce power in his own plant cheaper than he can purchase it elsewhere.

---

The Institute of Operating Engineers has the support and indorsement of able and clear-thinking men. If you are an operating engineer or hope ever to be one it might be worth your while to investigate it. The secretary's office is in the Engineering Societies building, New York City. He answers questions.

# Inquiries of General Interest

*Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it*

## *Flywheel Proportions*

I have a side-crank engine 11½x16 inches which is direct coupled to a fan and which I wish to reinstall to drive a laundry requiring 45 horsepower. The engine is designed to run at 120 revolutions per minute but can be adjusted to run faster. The steam pressure is 75 pounds.

I have two wheels 66 inches in diameter, 15-inch face, weighing about 1300 pounds, and would like if possible to use one or both of them. I have applied several formulas found in handbooks and none of them agree within several thousand pounds. The speed variation is not important and it would be possible to run the engine at a higher speed than stated to permit a lighter wheel.

L. C. R.

Flywheel formulas vary according to the conditions under which the wheel is to be used, and which must be known before the proper one can be selected. A flywheel suitable for a plain slide-valve engine driving a stone crusher might be only quarter heavy enough for a Corliss engine of the same power in an electric-lighting plant. For a plain slide-valve engine doing ordinary work, use the formula

$$W = 350{,}000 \frac{d^2 S}{D^2 R^2}$$

in which

$d$ = Diameter of the cylinder in inches;
$S$ = Stroke of the piston in inches;
$D$ = Diameter of the wheel in feet;
$R$ = Number of revolutions per minute;
$W$ = Weight of the wheel in pounds.

Substituting the numerical values in the equation it reads

$$\frac{11.5^2 \times 16}{5.5^2 \times 120^2} \times 350{,}000 = 1700 \text{ pounds}$$

the weight required in a flywheel 66 inches in diameter. If the wheels weigh 1300 pounds each, both will fill the bill very nicely or a slight increase in the number of revolutions will make one suitable.

## *Constant-current Transformer Working*

How can I tell when a constant-current transformer is overloaded? Why do the secondary windings break down if left on open circuit?

T. F.

If it regulates properly, a constant-current transformer cannot be overloaded in the secondary windings because the secondary current remains practically constant at all loads. The primary winding can be overloaded by increasing the resistance in the secondary circuit a good deal beyond that for which it was intended; this will cause the current in the primary winding to increase beyond the rated current. When the secondary circuit is opened, an excessive potential is induced in the secondary winding because the transformer automatically increases the secondary e.m.f. as the secondary resistance increases, in order to keep the current constant. The excessive e.m.f. caused by opening the secondary circuit breaks down the insulation.

## *Effect of Low Shaft*

What effect, if any, will a shaft 11/16 inch low from a line drawn through the center of the cylinder and guides have on a simple engine running over?

J. M. P.

On the out stroke the crank pin would travel over more than one-half of the crank-pin circle while on the return stroke it would travel over less than one-half of the circle. The effect would not be appreciable, or even measurable by any ordinary methods.

## *Motor Balancer Functions*

When two machines with coupled shafts are used to balance a three-wire direct-current circuit, do both work as motors or as dynamos, or is one a motor and the other a dynamo?

A. B. J.

With an unbalanced system, one machine acts as a motor and the other as a dynamo. Which is which depends on the load condition; the armature connected to the lightly loaded side of the system runs as a motor and the other one as a dynamo, these functions changing automatically as the unbalancing changes. When the two sides of the system are equally loaded, both machines run as motors.

## *Bad Speed Regulation*

A 250-kilowatt and a 125-kilowatt generator driven by separate engines run in parallel without difficulty but when the large machine is cut out, leaving the smaller one to carry the load, the voltage drops from 125 to about 75 or 80 volts for a few moments and then comes up to normal. When the small machine is disconnected from the load, its voltage runs up beyond the range of the voltmeter. What is the cause and how can it be removed?

W. H. L.

The cause is a sticky governor on the small engine. It seems evident that it is a shaft governor and the bar does not swing freely on the pivot pin. The remedy is obvious.

## *Constant-current Transformers*

Why are constant-current transformers used to connect a circuit of arc lamps to a primary alternating-current circuit? Could not 30 or 35 inclosed arcs requiring 65 volts each be connected in series directly across a 2300-volt primary circuit without using a transformer?

W. H. K.

For two reasons: First, in order to put any desired number of lamps in one circuit regardless of the voltage of the main supply circuit; second, in order to obtain regulation of the arc-lamp current. Thirty-four 65-volt lamps could be supplied directly from the 2300-volt circuit if the current would remain constant through the arcs, but by interposing a transformer any number of lamps can be supplied in a single circuit and the current will be kept practically constant by the transformer, no matter whether the lamps regulate accurately or not or whether they are all burning or some of them are cut out.

## *Direct-current Motor on Alternating-current Circuit*

What changes in connections are necessary to make a small series-wound direct-current fan motor work on an alternating-current circuit of the same voltage?

K. K. T.

Changing the connections will not produce the desired result. If the motor will not run on the alternating-current circuit as it is, which is very probable, it will have to be rewound. Instructions for rewinding cannot be given without complete knowledge as to the dimensions and magnetic character of all parts of the magnetic circuit of the machine.

# Refrigeration Department

*Principles and operation of ice making and refrigerating plant and machinery*

## Double Pipe vs. Atmospheric Condensers

By R. P. Kehoe

The popular favor with which the double-pipe type of condenser was accepted, was not due to increased efficiency nor cheaper first cost, but because it was something new and it looked nice. The water was not visible; no pan was necessary; and the condensers could be placed anywhere in a building. Furthermore, the water enters at one end and the ammonia at the other, introducing the countercurrent principle and thus promising extraordinary results.

It was these facts which gave the chief impetus to the sale of double-pipe condensers, and while there is no question about their advantages under a few favorable conditions, it is usually a mistake to adopt them on account of the disadvantages which are herein pointed out.

### Faults of Double-pipe Condensers

One of the worst features of this type is its inability to handle river water that is not absolutely free from vegetable matter or sewage. If only a small percentage of such impurities are contained in the water they will quickly collect in the fittings and pipes, causing a rapid decrease in efficiency and final stoppage if the pipes are not frequently cleaned.

A formation of the same thickness of scale on the atmospheric condenser and in the double-pipe condenser is more serious in the latter case. The scale in the atmospheric condenser gathers on the outer surface of the pipe and consequently increases the cooling surface, while the scale in a double-pipe condenser forms on the inner surface of the water pipe and decreases the cooling surface. Furthermore, less scale is required to form a certain thickness of deposit on the inner surface than on the outer surface of a pipe. The decrease of the transverse area of the water pipe from scale in a double-pipe condenser requires an increase in power to circulate a given quantity of water through it, or else if more power is not available, the quantity of cooling water is decreased. In an open-air ammonia condenser cooling water simply overflows the slotted-pipe gutter, and whether the scale is heavy or light on the pipes, the free flow of water either to the distributing gutter or over the condensing surface is not affected.

With the atmospheric condenser any deposit on the pipes is quickly perceived and may be readily scraped off, even while the condenser is in operation, whereas the double-pipe type must be discontinued from service while it is being cleaned.

In a cooling tower advantage is taken of the reduction in temperature resulting from the evaporation of some of the water to be cooled by a natural or forced air current. The more the evaporation the greater is the amount of heat carried off in this way. This principle is employed in an atmospheric condenser and figures largely in its high efficiency. This advantage is lost in the double-pipe type.

Also, in view of the great affinity of anhydrous ammonia for water, a leak in a double-pipe condenser may remain undiscovered for a long time.

During a recent winter, the engineer of a large brewery failed to drain all the water from a battery of double-pipe condensers when the plant was idle, and the water froze, resulting in many split pipes and fittings, and making it necessary to practically rebuild the condensers.

This type of condenser is usually placed inside the building, while atmospheric condensers are installed outside. In this respect, the latter naturally secure an advantage from the cooling effect during cold weather which is almost entirely lost by the former.

The efficiency of the double-pipe type can be maintained only by forcing the condensing water through the pipes at a fairly rapid rate, and due to the friction in the pipes a large amount of power is required for this purpose.

### First Cost

The comparative first cost of the two types f.o.b. cars at the factory is approximately as follows:

| | Atmospheric Type | Double-pipe Type |
|---|---|---|
| Diameter of pipes, inches | 2 | 1¼ and 2 |
| Number of pipes in hight of standard condensers | 24 | 12 |
| Length in feet | 20 | 18 |
| Square feet of cooling surface | 390 | 80 |
| Approximate cost of one condenser | $150 | $150 |
| Cost per square foot | $0.50 | $1.87 |

From these figures it will be seen that the first cost of the double-pipe condenser is nearly four times the cost of the atmospheric style per square foot of cooling surface.

The number and size of sections of both types usually furnished for each 100 tons refrigerating capacity per 24 hours are as follows:

| | Atmospheric Type | Double-pipe Type |
|---|---|---|
| Number of sections | 8 | 8 |
| Number of pipes in hight | 24 | 12 |
| Length in feet | 20 | 18 |
| Total cooling surface, square feet | 2400 | 640 |
| Total cost | $1200 | $1200 |

### Comparative Efficiency

In spite of the many faults previously mentioned, the double-pipe condenser has a high efficiency when operated under favorable conditions, such as good water, clean pipes and a high velocity of the condensing water. Good water, however, is available only in certain places; clean pipes are seldom found except in new plants and the velocity is naturally limited by a reasonable amount of power for pumping and the use of a reasonable amount of condensing water.

Giving the double-tube type the benefit of the most favorable conditions in practical operation, the comparative efficiency of the two types expressed in the number of B.t.u. exchanged per square foot of cooling surface per hour is as follows: Atmospheric, 60 B.t.u. per degree difference; double pipe, 100 B.t.u. per degree difference, with cooling water flowing at 250 feet per minute.

This greater efficiency makes it possible to use about 40 per cent. less surface but this is offset by the fact that the cost per square foot is increased nearly 300 per cent. Furthermore, in the average plant this efficiency would not be maintained because the tubes are not kept clean enough.

### Advantages of Double-pipe Condensers

In small plants up to 10 or 15 tons capacity it is often advantageous to use double-pipe condensers. They are small and compact, can be located close to the machine and, in view of the saving of connections and a condenser pan, are cheaper than the atmospheric style. Furthermore, when using city water that must be paid for, its merit is apparent, as no dirt is accumulated and the water consumption can be reduced to a minimum. It also has a field where the water is used again for other purposes.

After fairly summing up the advantages and disadvantages it is apparent that double-pipe condensers are not the best type to adopt in the usual refrigerating or ice-making plant except under special conditions. The practical faults alone should be sufficient to condemn them especially in ice plants, where all the apparatus is subjected to rough usage and where great cleanliness is not often practised. The simplest and most accessible apparatus should be preferred and while open-air condensers are not as pretty to look at, their simplicity is unquestionable.

There are several designs of the latter type which will be discussed.

### Atmospheric Types

The cheapest design of condenser is that in which the hot ammonia gas is led into the highest pipe in each stand and the liquid drawn off from the lowest pipe. It is also the least efficient not only per square foot of surface but in the head pressure maintained. The term "cheapest design," however, must not be misunderstood; it merely refers to the cost per square foot of cooling surface.

One of the most important considerations in the efficiency of a condenser is the head pressure. A few pounds difference in this pressure against which the compressor must work may mean hundreds of dollars every year either saved or thrown away. The heat transmission per square foot affects only the first cost and a few dollars more or less in the initial expense is not as vital as a loss that might go on year after year through high head pressure.

The pressure in a condenser is consistent with the temperature at which the ammonia begins to liquefy. The object, therefore, in an efficient design is to make this temperature as low as possible. With the type in which the gas enters the top pipe the condensing water is at first heated by the absorption of the superheat in the hot gas coming direct from the compressor. It is only after this heat has been removed that the ammonia can be brought down to the point of liquefaction, but then the temperature of the water has risen considerably and the ammonia must liquefy at a relatively high temperature. The result is a higher pressure than would be attained if the initial temperature of the water had been brought to bear on the ammonia at the point of liquefaction.

Another type of condenser is constructed with the gas entering the lowest pipe, from where it ascends through four pipes and then passes through a standpipe to the highest pipe. The ammonia then proceeds downward and the liquid is led off from the fifth pipe from the bottom. The theory upon which this condenser is based is that the four lower pipes (called the forecooler) will remove all the superheat and bring the gas very nearly to the point of liquefaction before it enters the top pipe, where the gas encounters the coldest water and then begins to liquefy at the lowest possible temperature. The practical faults of this type are that the four lower pipes, or the forecooler, cannot be depended upon to just remove the superheat.

Consider the difference in the amount of gas to be condensed according to the capacity developed; the difference in the temperature of the condensing water during all seasons of the year; and the variations in superheat of the gas in every plant. As these upset the theory unquestionably, it might be well to inquire into the result.

If all the superheat is not removed, the balance must be taken out after the gas has risen to the top pipe, and the same fault then exists to some extent as when the gas enters directly into this pipe from the compressor. The operation is much worse, however, if the conditions produce liquid in these lower pipes, which remains pocketed and interferes with the flow of gas until conditions change and it is regasified. To partly obviate this fault some designs provide a small connection from the lower pipes or precooler to drain off the liquid which may form. This special drain pipe is connected to the liquid main.

Reverting to the first principle of this design, it was the object of the precooler to deliver the gas to the top pipe at or near the point of liquefaction, so that it would liquefy at the lowest possible temperature. The introduction of the special liquid drain admits the fault of liquid forming in the lower pipes where the water is at the highest temperature. As a result, this liquid must govern the head pressure because liquefaction takes place here at the highest temperature. Therefore, the primary object of the design is entirely lost, as the head pressure corresponds with the temperature of the water leaving the condenser and not with the initial temperature. Under these conditions the precooler condenser will not do as well as the concurrent condenser.

### Countercurrent Atmospheric Condensers

As already pointed out, the double-pipe condenser owes its efficiency entirely to the fact that it is countercurrent in principle and may bring the temperature of the liquefying ammonia as nearly as possible to the initial temperature of the water. A countercurrent atmospheric condenser will therefore offer the same efficiency as the double-pipe type without the practical faults of the latter.

The countercurrent open-air condenser is arranged with the gas entering the lowest pipe and the liquid leaving through several drains at various levels. The superheat is removed in the lower pipes as in the precooler type and as the gas ascends it encounters the effect of colder and colder water. There is no space wasted by the accumulating liquid, which flows off as fast as it forms.

Such a condenser will meet every condition and can always be depended upon to give the best results. It never should be over 18 pipes in hight, as more than this are superfluous on account of the efficiency of the countercurrent effect. In fact the hight may be less. If the initial temperature of the water is high, it will be an advantage to make the stands only 12 or 14 pipes high and provide a greater number. More water is used if this is done, but the head pressure is lower because of the greater amount of work accomplished by the water.

Not long ago an experiment along these lines was attempted by the chief engineer of a large brewery. The plant was equipped with several countercurrent atmospheric condensers, each 24 pipes in hight; 12 pipes were removed from eve stand and the condenser pressure was found to be exactly the same. Later on the pipes which were removed were reërected in the form of additional stands and as soon as these new stands were put in operation the pressure dropped from 10 to 20 pounds.

A comparative test of the concurrent atmospheric condenser, with the gas entering the top pipe, and the countercurrent type, just described, will show that the latter can produce the same head pressure with nearly 20 per cent. less condensing water. If the same amount of condensing water is used, the head pressure will be reduced 10 pounds.

Hence, the countercurrent style will afford greater economy under all conditions. If water is scarce it can produce good results with a smaller quantity and if there is plenty of water available, it will reduce the head pressure and cut down the coal bill.

Yet it will be found that many of the large packing houses throughout the country use the open-air condenser of concurrent design almost exclusively. Some of these plants have one hundred or more condensers, all with the gas entering the top pipe, using more water and maintaining a higher head pressure than necessary. The only reason for this is that the operators have become accustomed to this style and do not like to change.

---

A recent patent describes a direct method for the production of ammonia by the passage of a mixture of hydrogen and nitrogen over finely divided osmium or uranium, or other similar catalytic material. The operation is conducted under about 1500 pounds pressure, and the ammonia is separated from the uncombined gases by cold.

# LETTERS

## Cutting Packing Over a Wooden Mandrel

In the April 25 issue selling section, I read with interest the talk on packing, by "Old Bill," of the Thermoid Rubber Company's advertisement. The wooden mandrel is all right and there should be several in every engine room, one for each machine, as the mandrel saves time and money.

In the plant in which I am working we have two ammonia piston rods of different sizes; one is 1/16 inch smaller than the other. I got a mandrel made

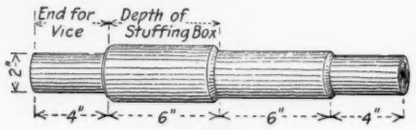


Fig 1

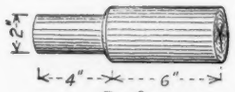


Fig. 2 POWER

out of hardwood, like the accompanying Fig. 1, for the ammonia piston rods and one like Fig. 2 for the steam piston rod.

The smaller ends of the mandrel shown in Fig. 1 will serve to put in the vise, and also the small end of Fig. 2. The enlarged parts may be as long as the stuffing boxes are deep. The diameter is to be 1/32 inch less than the diameter of the piston rod, so that there will be some space between the ends of the ring, when it is cut and placed over the rod. When the packing is warmed up, expansion will bring the ends together. In case the packing fits tight in the stuffing box, the space between the ends should be a little more.

The mandrel may be marked, for example, "No. 1 Machine; depth of stuffing box, 8 inches; nine rings of ¾-inch spiral." When cutting the packing, put the mandrel in the vise, take a thin wire nail and fasten the end of the spiral packing to one end of the mandrel, wind the former around the mandrel until it is all covered and fasten it with another nail at the end of the last ring. The packing may be cut straight across or on a slant.

For pumps the mandrel is very handy. It may be made to suit the size of the plunger, piston rod and valve stem. When packing the plunger with duck packing, more space must be allowed between the ends, and the follower plate should not be screwed up too high. The mandrel can be made in a short time by any engineer who has a lathe and knows how to use it.

WILLIAM L. KEIL.

Philadelphia, Penn.

## Air in Ice Water System

It is my opinion that Mr. Johnson's suggestion to prevent air in his ice-water system will not remedy the trouble. On account of the hight of the building, say 100 to 110 feet, and the design of the piping system, air will be drawn into the piping under certain conditions of operation. No means is provided to rid the system of this air and it is churned around in the centrifugal pump with the water, causing the latter to become milky. My suggestions would be: First, to get rid of the centrifugal pump and put in a triplex pump. Second, to discharge the cold water into a tank as high above the main on the tenth floor as possible and feed the system from this tank. A float should be provided in this tank to regulate the supply of fresh water taken in from the supply tank. The cold-water supply tank should be made flat like a pan and insulated. This will give a large disengaging surface for the air.

FRED OPHULS.

New York City.

## Opening an Ammonia Joint

At a certain plant my assistant engineer was instructed to break an ammonia joint. It was necessary to take out an ell and replace it with a tee and flanged valve so as to extend the coils in the cooling rooms. He claimed that the coils had been thoroughly pumped out.

I gave him a helper and he took all of the bolts out of the flanged ell and was prying the joint apart with a small bar, when it let go with a loud explosion and a shower of oil and muddy substances. A yell from the engineer followed and the next thing I knew he was all in a heap on the floor, choking with ammonia. I grabbed him and made for the open door.

The room was full of the fumes of ammonia and a hissing sound denoted a serious leak. I rushed for the valves in the engine room to isolate that coil and found one valve partly open. None of us was ever able to account for that open valve. My assistant had mistaken the valves and pumped out the wrong coil.

I started the machine and pumped some of the ammonia from the coil into the system, but soon stopped as I did not want to get air into the system.

As the engineer's lips, eyes and tongue were badly burned by the ammonia, he was taught a lesson which he will not forget.

The moral is, never remove all of the bolts on a joint at one time, but loosen them all a little and break the joint gently; then in case of a serious loss of ammonia, the bolts can be tightened up with safety and an investigation made.

D. L. FAGNAN.

New York City.

## Clearance in Compressors

In designing ammonia compressors it has been the practice of most of the manufacturers to cut down compressor clearance to the smallest possible degree in order to obtain what is commonly believed to be the maximum capacity per cubic foot of compressor displacement. Several other factors, however, should be considered in this connection which tend to militate against the successful working of a compressor under the above conditions. Prominent among these are safety of operation and the superheating of suction gas as it enters the compressor.

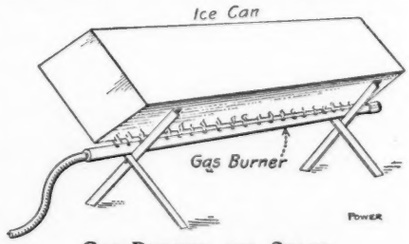
Dwelshauvers-Dery has demonstrated that the cylinder walls of a compressor have a very considerable thermal influence upon the working medium, which influence depends in amount on the conditions of operation and not upon the size of the compressor. It is a well known fact that, although clearance reduces the apparent volumetric efficiency, the horsepower necessary to compress 1 cubic foot of gas in a compressor with reasonable clearance remains the same as in a compressor without clearance. The reason for this is that the cylinder walls of a no-clearance compressor will superheat the gas more, and thereby reduce the capacity to a greater extent, than will a compressor with clearance reduce the apparent capacity. It is the effect of cylinder superheating which must be fully understood before the proper size of the clearance can be determined upon in the design of a compressor.

E. A. MURPHY.

New York City.

## Temporary Can Repair Kink

The following method was employed by an ingenious engineer to solder leaky cams. It is inexpensive and has prolonged the lives of cans as much as two



GAS BURNER AND STAND

years. The materials used are an equal mixture of turpentiine and beeswax.

To apply this mixture a gas burner made from a 1-inch pipe the same length as the can to be repaired is used. The pipe is perforated with small holes and mounted on a rack which is also built to hold a can in such a position that the corner will be directly over the jets. The mixture is poured in hot so that it will penertate the smallest crevices.

EDWARD T. BINNS.

Philadelphia, Penn.

# New Power House Equipment

*What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news*

## Improved Flow Meters

The General Electric Company, Schenectady, N. Y., has developed several types of flow meter of approved design and high efficiency. These include both the recording and indicating types, the former making a continuous graphic record of the rate of flow and the latter giving readings of instantaneous values of the same. The unit of measurement varies with the commodity measured, being pounds per hour for steam, cubic feet per minute for air and gas and gallons per minute for water.

The recording water-flow meter, shown in Fig. 1, comprises a nozzle plug for screwing into the pipe at the point where the flow is to be measured, thus being exposed to the pressure of the water, a meter element which measures the pressure set up in the nozzle plug and a recording mechanism which makes a graphic record of the rate of flow.

The nozzle plug, shown in Fig. 2, has two sets of orifices: the leading set, located parallel to its axis extending across the main and facing the direction of flow, and the trailing set, consisting of three holes located on the opposite side of the plug near the middle and at right angles to its axis. The two sets of orifices open into separate longitudinal chambers leading to the outer end of the plug. The nozzle plug is arranged for use on either vertical or horizontal mains and its introduction is said to cause no appreciable drop in static pressure even at very high rates of flow.

When the plug is inserted in the main, the water impinging against the orifices of the leading set develops therein a pressure equal to the static pressure plus a pressure due to the velocity head; while the pressure set up in the orifices of the trailing set is equal to the static pressure minus the pressure due to the velocity head, thus causing different pressures in the longitudinal chambers leading to the outer end of the plug. The different pressures are communicated to the meter through pipes attached to the outer end of the plug and connected to flexible steel tubing placed within the meter case.

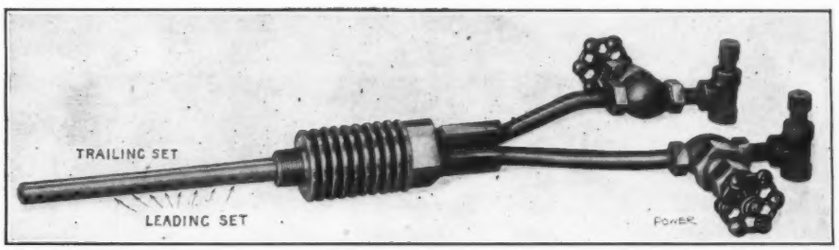



FIG. 2. NOZZLE PLUG AND PIPING FOR WATER-FLOW METER

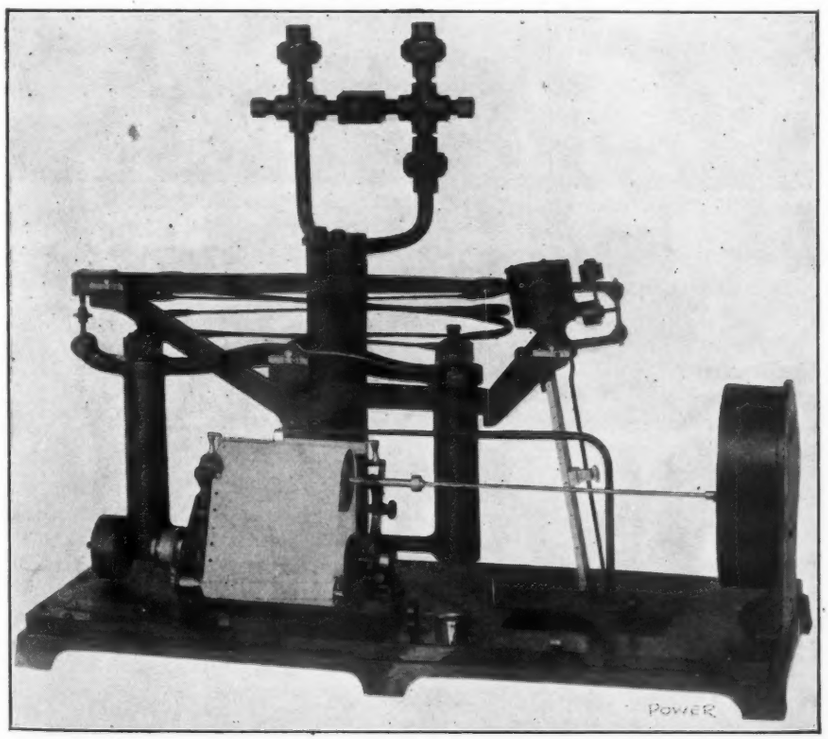



FIG. 1. RECORDING WATER-FLOW METER WITH AUTOMATIC-PRESSURE COMPENSATION

The meter element consists of two vertical cups or hollow cylinders connected at the bottom by a tube. The U-tube thus formed is filled with mercury to about half the hight of the vertical cups and the whole arrangement is supported and is free to move about a set of knife edges like an accurately constructed balance.

The different pressures in the two sets of orifices in the nozzle plug cause the mercury to rise in the left-hand cup and to fall a corresponding amount in the right-hand cup until the unbalanced mercurial column exactly balances the difference in pressure. This action causes the beam carrying the cups to descend on the left-hand side of the knife edges until the moment of the weights on the right-hand side exactly balances the moment caused by the displacement of the mercury into the left-hand cup. The motion of the beam is multiplied by means of levers and actuates the recording pen, the movement of the latter being proportional to the amount of mercury displaced.

The drum carrying the recording paper is driven by an eight-day clock. The paper is calibrated to record the flow in gallons per minute and the rate of flow may be read at any instant, or the average rate calculated for any given length of time.

The standard paper furnished is calibrated for a rate of feed of 3 inches per hour, but paper for feeding at a rate of 1 inch or 6 inches per hour can be supplied if desired. The meter is equipped with a spring-operated reroll device capable of holding one complete roll of paper.

Although the meter is calibrated to record the rate of flow in gallons per minute at 39.1 degrees Fahrenheit, suitable means are provided for readily setting it for different temperatures, pipe diameters and rates of flow. In order to measure the flow under normal conditions in any number of different pipes, it is only necessary to use nozzle plugs of sufficient length to extend across the pipes and record paper of suitable calibration range.

This meter is useful for ascertaining the output of pumping plants, the total amount of water consumed by a municipality or the amount distributed to different sections thereof, the input to water turbines and their loss of efficiency, the amount of water consumed in manufacturing processes, the amount of feed water delivered to boilers, the amount of cooling water used in condensers, the slippage in pumps due to leaky plunger packing or worn-out valves, and for discovering losses due to leaks in water mains.

FIG. 3. INDICATING STEAM-FLOW METER, SHOWING SCALES

The recording air-flow meter, suitable also for measuring steam and gas, operates on the same principle as the recording water-flow meter and is practically similar in all details of construction.

The indicating steam-flow meter, shown in Fig. 3, is designed for testing work and other purposes, such as locating trouble due to leaks, determining efficiency of boilers, etc., where accurate, instantaneous readings of the rate of flow are desirable.

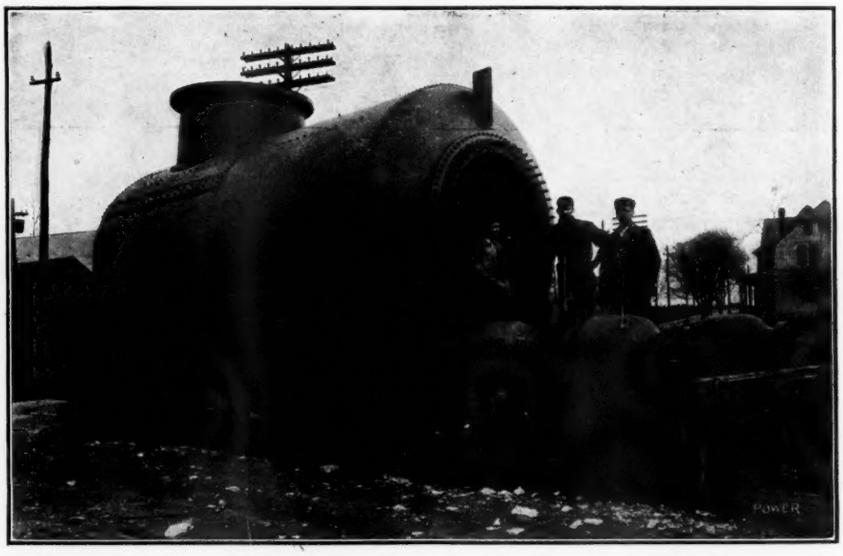
This type of meter operates on the same principle and employs the same type of nozzle plug as the recording meter, but differs considerably in details of construction. It can be readily adjusted for various rates of flow and indicates the instantaneous rate of flow in pounds per square inch of pipe cross-sectional area for steam, or in cubic feet of free air at 70 degrees Fahrenheit, for air.

## An Oil Eliminator for 66-inch Exhaust Pipe

The accompanying illustration shows a steel shell oil eliminator of the right-angle type recently built by the Hoppes Manufacturing Company, of Springfield, O., for the Milwaukee Electric Railway and Light Company, Milwaukee, Wis.

This apparatus is designed to remove the oil and water from 385,000 pounds of exhaust steam per hour before it passes to and is used in the operation of two 7500-kilowatt low-pressure turbine units. It embodies all of the Hoppes principles of construction for this type of eliminator, having large internal areas for the free and unrestricted flow of the steam and being thoroughly equipped with intercepting troughs partly filled with water to catch and remove all entrainment. The cylindrical shell of this machine is 10 feet in diameter and 23 feet long and when in place will stand vertically. The exhaust steam from the reciprocating engines enters at the side near the top and the purified steam passes out at the bottom.

Some idea of the enormous capacity of this machine may be had from the fact that the exhaust inlet flange is 5 feet 6 inches in diameter, the eliminator being installed in an exhaust pipe line of like dimension. The large special tee at the bottom for connecting to the turbines is not shown in the photograph.




LARGE OIL ELIMINATOR IN COURSE OF SHIPMENT

## Boiler Explosion at Alton, Ill.

BY H. R. ROCKWELL

A serious accident recently occurred at the power plant of the Illinois Glass Company at Alton, Ill., in the shape of a rupture of a 66-inch by 18-foot horizontal tubular boiler. This boiler was one of a battery of four of the same class, all connected to one common 12-inch header. The shell plate was of ⅜-inch steel. The writer was unable to find any brand on the plate signifying its tensile strength, but from the appearance of the ruptured plate it seemed to be of first-class material. The longitudinal seams were of the double-strap butt type triple riveted and the boiler contained 54 four-inch tubes.

The boiler is composed of two 9-foot sheets. The front plate was split from end to end, and 43 rivets were sheared on the girth seam and 23 on the head. About one-third of the flues were pulled from the front head and several were jammed through the back head of the boiler, which was raised bodily from its setting and badly damaged a steel roof truss about 20 feet above it. One of the other three boilers was thrown out of its setting and the remaining two were twisted enough to break the 12 and 12 by 6-inch tees which connected them to the header.

Two men were seriously scalded and one of them has since died. The accident happened just at 6 a.m. as the night and day shifts were changing. This very probably accounts for no greater fatality.

The man who was in charge of the boilers claimed that about 15 minutes prior to the accident he blew down the boilers and walked along the front and blew out each water column. The water rose to its proper level in each boiler.

The writer had charge of the plant and upon his arrival, about one hour after the accident occurred, he found that the water-column connections on the exploded boiler were open and the blowoff valve was closed just as they should be.

My theory of the accident is that a large quantity of scale was precipitated from the shell and tubes of the boiler and settled on the lower sheet over the fire, which allowed the sheet to become overheated sufficiently to start the initial rupture. Some seem to think that the explosion was a case of no water, but the damage to the boiler was too great for me to think that there was not plenty of water in it at the time.

## Water Works Association Convention

About five hundred delegates and guests attended the thirty-first annual convention of the American Water Works Association at Rochester, N. Y., June 6 to 9. At 10 o'clock on Tuesday morning, Mayor Edgarton gave a cordial welcome to the association in the banquet hall of the Powers hotel, to which John W. Alvord, president of the association, responded.

Some seventy manufacturers of waterworks appliances were represented with exhibits which covered stands and lined the corridor and sample rooms. After the routine business was despatched, the presentation of papers relating directly to water works was begun.

### Pumping-station Management

Thomas McMillan read a paper on pumping-station management in which he traced the history of the Milwaukee, Wis., water works from 1872 down to the present, giving short descriptions of the additions made from time to time, coupled with statements of the cost of operation for each ten-year period. The methods followed in the purchase and handling of coal were minutely described, as were also the boilers and the details of their operation. He described the special arrangements in steam and water piping for facilitating repairs, feeding the boilers and for increasing the temperature of the feed water, the cost of the installation and the operation of the oiling system, the auxiliaries and appliances for making and handling repairs and the general manner in which the operating force was managed in the appointing and arrangement of the work.

### Panama Canal

Tuesday evening, Dabney H. Maury, one of the engineers appointed by President Taft from the American Engineering Society to make an investigation of conditions at the Panama canal, gave an illustrated lecture in which he spoke especially of the economy which is being practised in building the canal and of the sanitation methods instituted to make conditions such that the men employed are better fitted to work. He said that the canal will be finished much sooner than expected. On June 1, 1913, it will be practically complete, and on that date vessels of 26-foot draft will be able to pass through. These vessels will thoroughly test out the canal until January 1, 1915, when the formal opening to all vessels will be made.

### Emergency Intakes

W. P. Mason, speaking on the above subject, said that when the water supply of a municipality is changed it is the common practice to allow the old connections to remain as an emergency supply. Such practice may reduce insurance rates, but it increases typhoid risks, and the question becomes, which is better to lose, a house by fire or a life from fever?

Numerous instances were cited where the use of an emergency intake in the case of fire was followed by an epidemic of typhoid because of the polluted water which was pumped into the regular water mains.

### Steel and Cast-iron Water Pipes

This topic was discussed by Allen Hazen. When large steel plates became available their fitness for use in the manufacture of high-pressure water mains was investigated. Steel pipe 36 inches in diameter costs about two-thirds as much as cast-iron pipe of equal strength and is believed to be in most places equally lasting, while because it is malleable it is much more reliable in trenches where bending and deformation from settling or shifting are liable to occur.

With the hundreds of miles of steel pipe in service there is no record of a destructive failure, while a break in cast pipe is invariably destructive and the flow continues until the reservoir is empty or the water is shut off. Steel pipe has not been standardized, but the earlier installations were designed by able men who established good precedents.

Surrounding steel pipe with concrete adds to its stability and durability and increases its cost, but not to the extent that will equal cast iron.

### Fire-line Methods

In this paper, by George Houston, attention is called to the fact that many if not all of the insurance companies object to all types of water meters except one in fire lines on the ground that they catch trash.

Question was raised as to whether the particular meter named was not also a trash catcher in that it allowed débris to pass into the branches of a sprinkler system and choke the smaller pipes.

It was suggested that with the current type of meter on a sprinkler system any matter that would choke the smaller pipes would be caught by the meter, and even though it stopped the rotation of the propeller blades it would still allow enough water to pass to extinguish an incipient fire, while with the favored meter the matter that would choke the sprinklers would pass the meter only to stop the flow where it was most needed. Citation was made of a specific case in which there was no meter in the line and the pipe leading to a single room was choked as soon as a few sprinklers opened, causing a property loss of about $90,000.

### Hot-water Troubles

George C. Whipple read a paper on the troubles that come from rusty hot water, due to the corrosion of iron and steel service pipes, tanks and boilers. The list of natural waters which cause corrosion included nearly if not quite all not distinctly alkaline. No one remedy is suitable in all cases, but, where practicable, lime or soda ash and the use of appropriate materials in pipes and waters is advised.

### The Purification of Drinking Water

The above was the subject of a paper by John L. Leal. As about 70 per cent. of the weight of the human body is water the importance of wholesome water is evident. It should not contain either organic or inorganic matter enough to interfere with its solvent properties, nor should it carry specific germs of disease.

Rain water, when it reaches the earth, is already fouled by impurities taken from the air, and in soaking into the ground is still further polluted by bacteria and soluble matter in the upper layers of soil. Filtration, which purifies it to some degree, takes place as it passes through the pervious soil to some impenetrable stratum along which it flows until it reaches its lowest level, where it is held or finds its way to the surface at some point below the ground level. Such water is usually of a high degree of purity, though due to a loose, shallow soil and crevices, springs which are fed by what is practically surface water may supply water impregnated with typhoid germs.

Sedimentation, sand filtration, mechanical filtration with coagulation, oxonation and sterilization or disinfection have all been successfully adopted as means of water purification.

Sedimentation is an old method practised in ancient Babylon and Egypt. Sand filtration dates from 1829. In mechanical filtration and coagulation the water is usually first treated with sulphate of alumina and then filtered through gravel and sand.

There are upward of 600 plants of this type in operation in the United States, and there seems to be little doubt

that it is the most suitable practical system for water purification.

Oxonation has never been practically tested on a large scale, and sterilization has had only a limited application, the first time it was ever used in a continuous process being at Boonton, N. J., in 1908. However, the sterilization process marks a great step in advance on account of cheapness, simplicity, efficiency and certainty.

## Wisconsin N. A. S. E. Convention

Eleven years ago Milwaukee was the scene of the national convention of the National Association of Stationary Engineers. During those eleven years there has been wonderful progress in the growth of the organization and it was freely said by those who had attended the national meeting that the late State meeting held there June 8, 9 and 10 was larger and better in many ways.

H. J. Mistele, chairman of the local committee, presided at the opening exercises and introduced Mayor Emil Seidel, who delivered the address of welcome. The response was by State President A. A. Schroeder, of La Crosse, following which Fred W. Raven, national secretary, spoke on the "National Association of Stationary Engineers."

E. P. Gould, secretary of the Central States Exhibitors' Association, was also called on for a few remarks regarding the business end of convention work, and R. D. Tomlinson, of the Allis-Chalmers Company, who is a past national president of the organization, gave a short address.

Educational work was prominent during the convention. John W. Lane, editor of the *National Engineer*, in an address on the subject, gave many valuable suggestions for increasing the value of this department to the organization.

"Heat and Ventilation" was the subject of a lecture by B. J. Miller, of Milwaukee, some of the fundamental principles of this subject being taken up and explained in detail. Another lecture, that on the "Nonreturn Stop Valve," by E. P. Gould, of Chicago, contained a clear and concise statement of what this important piece of equipment should be and of some of the defects which are commonly found in its operation.

On Friday afternoon a visit was made to the plant of the Richardson-Phenix Company. The smoker given by the Central States Exhibitors' Association in the main dining room of the Plankington house on Saturday night was presided over by Royal D. Tomlinson, and proved a great success in promoting good fellowship and good cheer between the engineers and the supplymen.

On Sunday morning a baseball game between the engineers and supplymen was the principal event of interest; the game was won by the former by a score of 12 to 3.

Officers for the ensuing year were elected as follows: William Classman, of Milwaukee, president; Henry Holst, of La Crosse, vice-president; Robert Fenn, of Sheboygan, secretary; John Murphy, of Madison, treasurer; H. Breitbach, of Stevens Point, conductor, and Dan Dreger, of Manitowoc, doorkeeper. The place of next meeting will be decided by the State officers.

THE WISCONSIN DELEGATION AT STATE CONVENTION

Exhibitors at the convention were as follows: Allis-Chalmers Company, American Steam Gauge and Valve Manufacturing Company, J. Andrae & Sons Company, V. D. Anderson Company, Chase Brothers Company, Crandall Packing Company, George B. Carpenter Company, G. M. Davis Regulator Company, Dearborn Drug and Chemical Works, Garlock Packing Company, Greene, Tweed & Co., Philip Gross Hardware Company, Hawk-Eye Compound Company, Hills McCanna Company, Hoyer Metallic Packing Company, Jenkins Brothers, H. W. Johns-Manville Company, Keystone Lubricating Company, Lunkenheimer Company, Lyons Boiler Works, Mechanical Appliance Company, Milwaukee Factory Supplies Company, Michigan Lubricator Company, *National Engineer*, Osborne High-Pressure Joint and Valve Company, Perfection Heater and Purifier Company, William Powell Company, POWER, *Practical Engineer*, Richardson-Phenix Company, Scott Valve Company, Fred Sprinkman & Son, Steam Appliance Company, Swift Fuel Company, Vilter Manufacturing Company, Wadhams Oil Company, Western Iron Stores Company, Wickes Boiler Company.

## Identification of Power House Piping

The committee of the American Society of Mechanical Engineers on the identification of power-house piping recently turned in the following report:

In the main engine rooms of plants which are well lighted, and where the functions of the exposed pipes are obvious, all pipes shall be painted to conform to the color scheme of the room; and if it is desirable to distinguish pipe systems, colors shall be used only on flanges and on valve-fitting flanges.

In all other parts of the plant, such as boiler house, basements, etc., all pipes (exclusive of valves, flanges and fittings), except the fire system, shall be painted black, or some other single, plain, durable, inexpensive color.

All fire lines (suction and discharge), including pipe lines, valve flanges and fittings, shall be painted red throughout.

The edges of all flanges, fittings or valve flanges on pipe lines larger than 4 inches inside diameter, and the entire fittings, valves and flanges on lines 4 inches inside diameter and smaller, shall be painted the following distinguishing colors:

DISTINGUISHING COLORS TO BE USED ON VALVES, FLANGES AND FITTINGS ONLY

| | |
|---|---|
| Steam division | High pressure—white<br>Exhaust steam—buff |
| Water division | Fresh water, low pressure—blue<br>Fresh water, high pressure boiler feed lines blue and white<br>Salt water piping—green |
| Oil division | Delivery and discharge—brass or bronze yellow |
| Pneumatic division | All pipes — gray |
| Gas division | City lighting service aluminum<br>Gas engine service—black, red flanges |
| Fuel oil division | All piping—black |
| Refrigerating system | White and green stripes alternately on flanges and fittings, body of pipe black |
| Electric lines and feeders | Black and red stripes alternately on flanges and fittings, body of pipe being black |

# Moments with the Ad. Editor

*A department for subscribers edited by the advertising service department of Power*

[EDITOR'S NOTE—Here is an article that will interest engineers and manufacturers alike. It was written by a man who has had years of experience in the power-machinery field and will be published in two instalments, of which this is the first.]

## Who Has The Say?

BY C. A. TUPPER *

IN the selection of equipment for a power plant, does the operating engineer have much, if anything, to say?

Some years ago I thought not; today, I would answer that he does—very decidedly.

What has brought about the change in opinion? Merely experience.

Among manufacturers of machinery there are many, however, in places of authority, who would not be inclined to agree with me. With the increasing numbers and importance of consulting engineers, it has come to be a serious question in the minds of these officials how far the purchase of equipment is influenced by the views of the men who are to operate it. What, for example, has the engineer in charge of a power plant to say about the buying of a boiler, an engine, a feed-water heater or a pump?

I have heard sales managers, or other executives, declare that his influence has waned. "For an entirely new plant," they affirm, "the selection of equipment is governed chiefly by one of three things, viz: the personal preference of the owner or his manager; the recommendation of the consulting engineer, or the matter of price." It is admitted that, in adding to an existing plant or building another to replace it, the operating engineer then in the employ of the owner is likely to be consulted. "But, even then," say they, "his views do not carry the weight that they once did."

From this judgment I dissent. As one who has been through all branches of the business—manufacturing, advertising, selling and consulting work, I am led both by my own experience and by observation, to believe that the influence of the operating engineer is increasing rather than decreasing. The reason for it is fundamentally sound.

Today, operating engineers are men more thoroughly trained, as a rule, than those of bygone years. They understand the theory of their profession as well as its practical aspects, and they are capable of making comparisons and tests that put the service records of their machinery on a scientific basis.

In the case of the large central plant for electric power and lighting, where a complex system needs to be handled, the engineers in charge are now almost invariably men who have had a thoroughly technical education; and, for my part, I would rather intrust to such men the laying out of a new station, with all that implies, than to the average consulting engineer who is the graduate of a good university but weak on operating details through not having had their practical experience. What I have said of training applies not only to the engineers regularly employed by large power, lighting and traction companies engaged in public service, but also to the men in charge of the power systems of important industrial enterprises, including mills and factories of all kinds.

For the smaller plants equipped with modern machinery the training required to successfully operate them is also of a much better character than formerly; so that, altogether, the work of the operating engineer is on a higher plane today than ever before. With better training and increased responsibilities has come the more general acknowledgment of plant owners that the men to guide the selection of equipment are the men most conversant with its use.

Salesmen who ignore this fact—usually young salesmen—often wonder why they fail to secure certain contracts which were apparently within their grasp. They have made a good impression on the manager, who is possibly a friend or relative or old "college chum"; they feel sure that their prices are in line; but, for some reason, the sale has slipped away from them. In their report to the house they attribute this, as a general thing, to anything but the true cause. To illustrate:

A company manufacturing rubber goods wrote to one of the leading manufacturers of steam-plant specialties that it contemplated making changes in its feed-water heaters, pumps, etc., to secure greater economy. It would be glad to have someone, who was experienced in this particular line, call and make recommendations. A brilliant young sales engineer was picked for the mission. He speedily got on very good terms with the manager and superintendent, made a careful examination of conditions and departed, promising to send in a sketch showing the rearrangement, new equipment, etc., that would bring about the results they desired.

Meanwhile, he had talked with the engineer in charge of the plant; but his conversation, as he went along, was directed almost entirely to the superintendent, and his attitude toward the plant engineer was tinged with an air of superior knowledge which the latter resented.

In his talk he spoke of a new type of heater that his company was then putting on the market; but, when the local engineer asked for an explanation of some details that were not clear from what had been said, the salesman answered only briefly, with no apparent desire to enlist the engineer's interest. In fact, he treated him as a negligible factor, useful for giving information about the conditions in the plant, but of no consequence in the purchase of equipment.

Three days later the superintendent received a carefully executed sketch showing the rearrangement needed. Accompanying it was a list of the apparatus which the manufacturer proposed to furnish, with prices, etc. These were turned over to the operating engineer.

The latter commented favorably upon the plan, which was undoubtedly a clever one, but thought that a different type of heater, with which he was familiar, would give just as good results if installed in the same manner, and it could be purchased for considerably less. Accordingly, the superintendent was induced to call in the salesman of another company.

This salesman was a graduate not of any university but of the school of hard knocks. He knew every detail of power-plant service and was acquainted with every operating engineer in his territory. Wherever he went he consulted their preferences, took advantage of all the information that they had to give him concerning local conditions and got them so interested in his apparatus that they had a lasting conviction in relation to its good points. Other things being equal, he usually sold his apparatus. He placed it in this instance. With a few alterations, the plan submitted by the young sales engineer was adopted; but it did not include the equipment that he was employed to sell.

Now the pity of it was that the new type of heater would have repaid, many times over in the course of a year, the increase in the price asked for it. If its advantages, which were not then widely advertised as they are now, had been carefully explained to the engineer in charge of the plant, he would readily have seen them, and, with his indorsement, the sale could have been promptly closed.

The sales engineer reported to his manager that the contract had been lost on account of the higher price they were obliged to ask for superior apparatus. This was not true. He had merely failed to recognize the understanding and influence of a man who wore overalls and did not put on airs.

From my own recollection I could draw numerous similar instances. I might tell how a battery of very economical boilers was not purchased because the principle on which they were designed had not been properly explained to the engineer of a large plant; how a four-valve engine was chosen in place of a Corliss because the operating engineer had seen it advertised in the paper he read regularly and was interested enough to ask for further details; how a steam-turbine negotiation was introduced and won out at the eleventh hour because the engineer of the system, about to attend a convention of stationary engineers, had seen the announcement of a model to be exhibited there and requested further time to investigate this type of machine; or how certain grades of packing, lubricating oils, belting, etc., have gained acceptance practically as "standards" through intelligent advertising to the men who direct their use.

*(To be continued and concluded next week.)*

*Formerly with Allis-Chalmers Company.

# BUSINESS ITEMS

Recent transformer business secured by the Westinghouse Electric and Manufacturing Company, East Pittsburg, Penn., includes an order from the Desert Power and Water Company, of Kingman, Ariz., for four 600-kva. oil-insulated, water-cooled transformers.

The O. D. Havard Company, Scranton, Penn., reports the receipt of an order from the Delaware, Lackawanna & Western Railroad for twelve combination sets of hopper and coal meters for conveyer lines for its mines; also an order from the Delaware & Hudson Railroad for sixteen coal meters for 9-inch pipes.

The Engineers Appliance Company has been incorporated for the purpose of manufacturing and selling steam and water appliances and has taken over the valve business of the Cleveland Steel Tool Company, including the Hydromatic water-control valve. The sale of this valve will be pushed and new specialties will be brought out. The offices and plant are located at East Eighty-second street and Lake Shore & Michigan Southern Railway, Cleveland, Ohio.

# NEW EQUIPMENT

Lewiston, Me., will extend its water system.

Mars Hill, Me., will install a new water system.

Panama City, Fla., voted to issue bonds for waterworks.

Daytona, Fla., will vote on $8000 bonds for waterworks.

Middleton, N. S., will equip a new electric-lighting plant.

Stellarton, N. S., will install a new waterworks pumping plant.

Battleford, Sask., will install new waterworks pumping system.

Hamilton, Ont., will spend $500,000 on a new power and light plant.

Fire destroyed the power plant of the Bayway (N. J.) Chemical Company.

Cambridge, Mass., contemplates building a new municipal electric-lighting plant.

Harry C. Wilson, Lacrosse, Wash., is planning to erect a light and power plant.

The Springfield (Mass.) Provision Company will build a new cold-storage plant.

The city of Detroit, Mich., will install new boilers in its waterworks pumping station.

The installation of an electric-lighting system, at Custer, S. D., is being planned.

Canora, Sask., will spend $100,000 for new electric-lighting and waterworks machinery.

Fire caused $25,000 damage to the plant of the London (Ont.) Cold Storage Company.

Southington, Conn., has voted $220,000 bonds for improvements to its water system.

Northampton, Mass., is considering the establishment of a municipal lighting plant.

The city of Oakdale, Cal., has voted bonds for $82,000 for waterworks and sewer system.

Salem, Mass., will install a new high-pressure salt-water system. Pumps will be needed.

John Faulkner, Hutchinson, Kan., is planning the erection of an ice plant at Portales, N. M.

McComb, Miss., contemplates issuing $15,000 bonds for improving and extending waterworks.

The Otonabee Power Company, Peterboro, Ont., will buy $200,000 worth of new machinery.

The Indianapolis (Ind.) Water Company is building additions to its various plants in the city.

New machinery is to be purchased for the La Colle Falls power station, at Prince Albert, Sask.

Moncton, N. B., will buy $16,000 worth of additional machinery for waterworks pumping station.

Morgan City, La., contemplates the construction of waterworks and sewage system to cost $60,000.

The Central New England Railway Company will install boiler, engine and pumps at Maybrooke, N. Y.

The Adel Milling Company has been granted a franchise to install an electric-light system, at Dallas Center, Iowa.

J. C. Morton, Conconully, Wash., is planning to construct an electric-lighting plant and waterworks system.

The Santa Barbara (Cal.) Gas and Electric Company will issue $1,000,000 bonds for improvements and extensions.

The Schwarzschild & Sulzberger Packing Company will erect a meat-packing plant in Portland, Ore., to cost $500,000.

The Glens Falls (N. Y.) Portland Cement Company contemplates the installation of 600 horsepower additional boiler capacity.

The Great Falls (Mont.) Power Company is planning the erection of a series of substations between that place and Lewiston.

The Dunsmuir (Cal.) Water, Power and Light Company is planning improvements and additions to its plant to cost $75,000.

The Berkshire Street Railway Company is to erect a transformer station and also an addition to its power house in Pittsfield, Mass.

The city of Cleveland, Ohio, is planning the installation of a battery of water pumps for high-pressure fire service. A. B. Lea, director.

The Ripley Ice Manufacturing Company, Ripley, Ohio, has been organized with $10,000 capital by Ernst Baumbach, F. E. Ross, A. W. Francis, etc.

The Logan (Ohio) Light and Power Company has purchased the Citizens Electric Light Company, of Logan. Will install additional boilers.

Power-plant equipment will be required for the new five-story city hall to be erected at Spokane, Wash. Julius Zittel, Spokane, is preparing plans.

The city of Gainesville, Tex., purchased the waterworks plant of the Gainesville Water Company and will spend about $60,000 on improvements.

The Western States Gas and Electric Company, Eureka, Cal., has acquired the plant of the Ferndale (Cal.) Lighting Company. Will install new equipment.

The Navy Department, Bureau of Supplies and Accounts, Washington, D. C., will open bids June 27, for 16 forced-draft blowers as per Schedule No. 3684.

Franchise has been granted the Belleplaine (Minn.) Electric Light and Power Company to construct and operate an electric-light plant in Excelsior, Minn.

J. B. McCrary Company, Empire building, Atlanta, Ga., is preparing plans for waterworks for Colquitt, Hartwell, Lavonia, Tallapoosa and Villa Rica, Ga.

The Hydro-Electric Power Company, of Mt. Carmel, Ill., is planning the erection of a new power plant and dam to cost $600,000. W. H. Zimmerman, president.

The El Centro (Cal.) Cold Storage and Bottling Works, organized by W. C. and L. R. Thomas, will build a cold-storage plant and bottling plant on Third street.

The Thomas Devlin Manufacturing Company, Burlington, N. J., will increase the capacity of its power plant. New Corliss engine and generator will be installed.

The Bernheimer & Schwartz Pilsener Brewing Company, Amsterdam avenue and One Hundred and Twenty-eighth street, New York, will build an addition to its ice plant.

Power-plant and refrigerating equipment will be installed in the new hotel to be erected by W. J. Geierman, Altadena, Cal. Albert C. Martin, Los Angeles, architect.

Providence Hospital, Washington, D. C., has awarded contract for the erection of a power house and laundry. Henry Adams, Calvert building, Baltimore, consulting engineer.

Bids will be received July 1 for extensions to the municipal electric-light and waterworks systems at McKinzie, Tenn., for which $40,000 bonds were recently voted. C. H Jenks, engineer.

The city of Terrell, Tex., has awarded contract for the erection of a power house in which will be installed an electric-light plant and waterworks. H. Galbraith, chairman, water commission.

The Gorham Manufacturing Company, Providence, R. I., is installing additional 300-horsepower boilers and a new chimney. The work involves changes in steam piping, etc. C. E. Fairbanks, chief engineer, is in charge of work.

Proposals will be received until 11 a.m., July 8, at the Bureau of Yards and Docks, Navy Department, Washington, D. C., for auxiliary-plant equipment at the Mare Island, Cal., Navy Yard. Specifications can be had of R. C. Hollyday, chief of bureau.

W. L. Coulson, general manager, Canadian Collieries Company, at Comox, B. C., will buy a large quantity of hydroelectric equipment, including two direct-connected units of 3750 kilowatts each, and equipment for new power house for pipe line.

Proposals will be received until 11 a.m., July 8, at the Bureau of Yards and Docks, Navy Department, Washington, D. C., for boilers and auxiliary power-plant equipment for the United States naval station, at Pearl Harbor, Hawaii. Further information can be had of R. C. Hollyday, chief of bureau.

# HELP WANTED

*Advertisements under this head are inserted for 25 cents per line. About six words make a line.*

ENGINEERS to solicit orders for the Rolin patent adjustable grate bar. Apply Standard Grate Co., Heed Bldg., Philadelphia, Penn.

WANTED—Thoroughly competent steam specialty salesman: one that can sell high-grade goods. Address "M. M. Co.," POWER.

AN ENGINEER in each town to sell the best rocking grate for steam boilers. Write Martin Grate Co., 281 Dearborn Street, Chicago.

AGENTS WANTED for first-class steam specialty in use throughout United States. Address C. S. Wood, 410 S. 15th St., Philadelphia, Penn.

THE VULCAN SOOT CLEANER offers an exceptional opportunity for power specialty salesmen. For further particulars address

G. L. Simonds & Co., 802 Steinway Bldg., Chicago, Ill.

SALESMAN calling at power plants to handle as a side line superior packing for steam, different from the rest and better. Nugget Packing Co., 185 Summer St., Boston, Mass.

ESTIMATOR on steam power plant piping; must be thoroughly conversant with this particular line of work; steady position for right man; state salary desired. Address "Estimator," Box 454, POWER.

WANTED—An engineer in each city as agent for a high class water-back Scotch boiler, the most economical steam generator known to the trade. Kingsford Foundry & Machine Works, Oswego, N. Y.

DRAFTSMAN WANTED by a large pump manufacturer; a first-class man capable of working up new designs from rough sketches or verbal directions; must be a man with original ideas and initiative; for such a man we have a good opening where future advancement will be governed wholly by his abilities; give complete outline of experience, education, references, salary expected, etc. Box 455, POWER.

## SITUATIONS WANTED

*Advertisements under this head are inserted for 25 cents per line. About six words make a line.*

AS CHIEF in plant of 2500 horsepower, or under as assistant in larger; 26 years old, married; strictly temperate, own indicator, first class Ohio license, eight years' experience with all kinds of power plant machinery; satisfaction guaranteed; prefer West. Box 453, POWER.

TECHNICAL MAN, 34, married, 13 years' experience, power, light, telephone, construction and operation, want connection as electrical engineer or superintendent, light or power plant; good business ability; prefer West or South; would consider foreign country. Box 433, POWER.

ENGINEER, 16 years' experience, simple, compounds and tripple expansion engines, tubular and water tube boilers, six years oil burning furnaces, practical experience with indicators for economical results; last position eight years; high pressure water pumping station, 750 pounds per square inch; references if necessary. "Engineer," Box 586, Morrusi, Ariz.

## MISCELLANEOUS

*Advertisements under this head are inserted for 25 cents per line. About six words make a line.*

PATENTS. C. L. Parker, Patent Attorney, ex-examiner Patent Office, 904 G St., Washington, D. C. Write for Inventor's Handbook.

EVERY ENGINEER should be posted regarding the new system of vacuum heating installed without payment of royalty; I have valuable information; write today. T. L. Reeder, 1417 W. Jackson Blvd., Chicago, Ill.

WILL BUY slightly used machinery as follows, if in first class condition: One internal combustion engine, 200 to 300 horsepower, "Hornby-Akroyd preferred; one 24- to 36-inch centrifugal pump and 100 feet suction and discharge pipe for pump. Give price, f.o.b. cars, weights, description, freight rate to Mission, Texas, history of outfit with makers' numbers; only first class stuff, no junk wanted. Box 107, Mission, Texas.

## FOR SALE

*Advertisements under this head are inserted for 25 cents per line. About six words make a line.*

FOR SALE—Wain wright corrugated copper tube feed water heater . The St. John Wood Working Co., Stamford, Conn.

ASHCROFT STEAM GAGES, electric high and low alarm bell attachment complete, $2.00. Troy Electrical Co., Troy, N. Y.

FOR SALE—20x48 Wheelock engine and two 72"x18' high pressure tubular boilers in good condition cheap. Address "Engineer," Box 2, Station A, Cincinnati, Ohio.

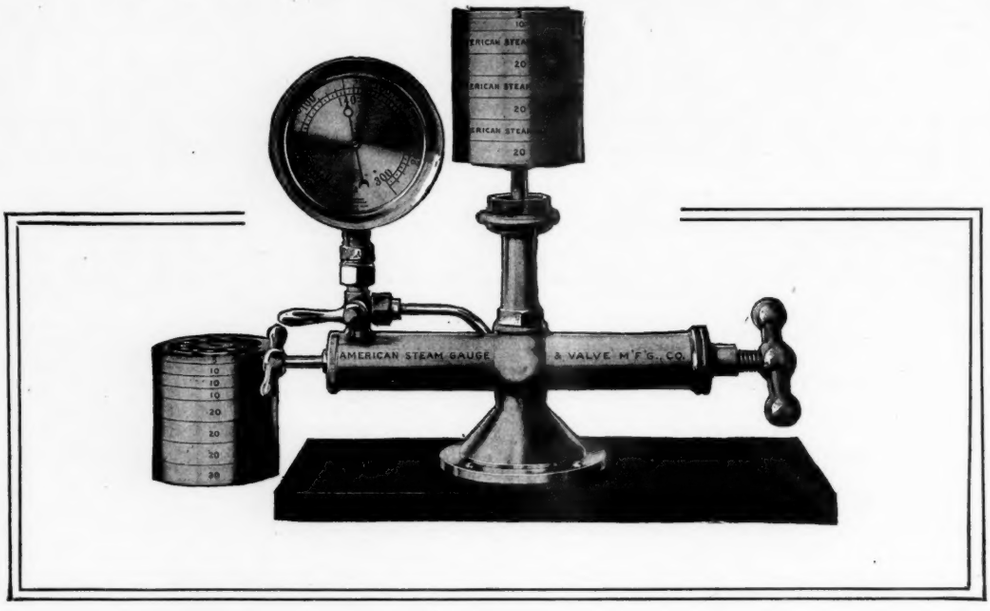
FOR SALE—For immediate delivery, two Cooper Corliss engines, direct connected to Westinghouse direct-current, 125-volt generators, 100 r.p.m., sizes 100 to 200 kilowatts; also one Westinghouse vertical direct connected unit 25 kilowatt capacity, 125 volts, direct current. Box 449, POWER.

FOR SALE—One compound Reynolds Corliss engine, high pressure cylinder 32 inches, low pressure 60 inches, stroke 60, 1200 horsepower at 100 pounds pressure, 70 revolutions per minute, flywheel diameter 20 feet, face 56 inches, built in sections; shaft length 16 feet diameter in wheel 16 inches, single eccentric. Box 434, POWER.

FOR SALE OR RENTAL—Factory with 100 horsepower suction gas producer (cheapest power). If rented, 30 to 60 horsepower available at $35 per horsepower. Rental space, 15 cents per square foot; total, available, 7000 square feet; insurance, $11.25 per thousand; water from deep driven well; location (between bridges) 16 and 20 Stuben street and 19 Grand avenue, Brooklyn, N. Y.; also yard room; will divide to suit; usual brokers' commissions allowed. A. B. Fisher, 81 Fulton Street, New York City, Telephone 3314 Beekman.

# ALPHABETICAL INDEX TO ADVERTISERS


| | PAGE |
|---|---|
| Air-Flex Co. | 108 |
| Albany Lubricating Co. | 1st cover |
| Alberger Condenser Co. | 89 |
| Alberger Gas Engine Co. | 113 |
| Allis-Chalmers Co. | 83 |
| American District Steam Co. | 111 |
| American Eng. Co. | 116 |
| American Mfg. Co. | 90 |
| American Metal Hose Co. | 82 |
| American Ship Windlass Co. | 101 |
| American Steam Gauge and Valve Mfg. Co. | 66 and 67 |
| Ames Iron Works | 114 |
| Anderson Co., V. D. | 88 |
| Appleby Co., Geo. | 85 |
| Armstrong Mfg. Co. | 94 |
| Ashton Valve Co. | 95 |
| Austin Separator Co. | 86 |
| Babcock & Wilcox Co. | 105 |
| Baker Valve Co. | 113 |
| Ball & Wood Co. | 116 |
| Ball Engine Co. | 116 |
| Baragwanath & Son, Wm. | 81 |
| Barnes Co., W. F. & John | 104 |
| Bayer Steam Soot Blower Co. | 106 |
| Beggs & Co., James | 84 |
| Betson Plastic Fire Brick Co. | 108 |
| Bignall & Keeler Mfg. Co. | 94 |
| Bogart Gas Power Engineering Co. | 113 |
| Borden Co. | 94 |
| Bowers Rubber Works | 99 |
| Bristol Co. | 4th cover |
| Bruce-Macbeth Engine Co. | 111 |
| Buckeye Engine Co. | 115 |
| Buffalo Forge Co. | 92 |
| Builders Iron Foundry | 108 |
| Burt Mfg. Co. | 18 |
| Cade & Knapp | 111 |
| Cameron Steam Pump Works, A. S. | 89 |
| Cancos Mfg. Co. | 98 |
| Casey-Hedges Co. | 135 |
| Chaplin-Fulton Co. | 106 |
| Chapman Valve Mfg. Co. | 96 |
| Chesterton Co., A. W. | 88 |
| Cling-Surface Co. | 90 |
| Columbian Rope Co. | 90 |
| Conard, Thos. P. | 76 |
| Cooper Co., C. & G. | 113 |
| Consolidation Coal Co. | 79 |
| Corliss Chemical Co. | 106 |
| Crosby Steam Gage & Valve Co. | 95 |
| Curtis & Curtis Co. | 95 |
| Dart Mfg. Co., E. M. | 94 |
| Deane Steam Pump Co. | 77 |
| Dearborn Drug & Chemical Works | 5 |
| Deming Co. | 89 |
| Detroit Lubricator Co. | 85 |
| Diamond Rubber Co. | 99 |
| Dixon Crucible Co., Jos. | 84 |
| Du Bois Iron Works | 113 |
| Elliott Co. | 7 |
| Epping-Carpenter Co. | 89 |
| Eureka Packing Co. | 74 |
| Fitchburg Steam Engine Co. | 115 |
| Flexible Steel Lacing Co., | 4th cover |
| Foos Gas Engine Co. | 113 |
| Garlock Packing Co. | 9 |
| General Electric Co. | 78 |
| Goulds Mfg. Co., The | 88 |
| Green Engineering Co. | 101 |
| Greene, Tweed & Co., | 85 and 4th cover |
| Griffith & Wedge Co. | 114 |
| Griscom-Spencer Co. | 80 |
| Hammel Oil Burner Co. | 83 |
| Harbison-Walker Refractories Co. | 109 |
| Harris Steam Engine Co., W. A. | 116 |
| Harrisburg Fdry. and Mach. Works | 116 |
| Harrison Safety Boiler Wks. | 69 |
| Heine Safety Boiler Co. | 104 |
| Hewes & Phillips Iron Wks. | 116 |
| Hoffman, Geo. W. | 88 |
| Holyoke Steam Boiler Works, Inc. | 107 |
| Homestead Valve Mfg. Co. | 96 |
| Hooven, Owens, Rentschler Co. | 115 |
| Hoppes Mfg. Co. | 102 |
| Hughson Steam Specialty Co. | 96 |
| Huyette Co., Inc., Paul B. | 102 |
| Ide & Sons, A. L. | 116 |
| Ingersoll-Rand Co. | 104 |
| International Corr. Schools | 73 |
| Jamieson & Co., M. W. | 83 |
| Jefferson Union Co. | 10 |
| Jenkins Bros. | 71 |
| Johns-Manville Co., H. W. | 83 |
| Johnson Rule Mfg. Co., E. P. | 83 |
| Keeler Co., E. | 104 |
| Kellogg Co., The M. W. | 94 |
| Kennicott Co. | 81 |
| Keystone Lubricating Co. | 3 |
| Kingsford Fdry. & Mach. Wks. | 105 |
| Kinkead Mfg. Co. | 82 |
| Lagonda Mfg. Co. | 3d cover |
| Liberty Mfg. Co. | 7 |
| Long Grate Bar Co. | 100 |
| Lunkenheimer Co. | 4 |
| Lytton Mfg. Corporation | 87 |
| McClave-Brooks Co. | 14 |
| McGraw-Hill Book Co. | 110 |
| McIntosh, Seymour & Co. | 116 |
| McLeod & Henry Co. | 108 |
| Machinery Brokerage Co. | 76 |
| Magnolia Metal Co. | 112 |
| Manzel Bros. Co. | 85 |
| Martin Grate Co. | 100 |
| Mason Regulator Co. | 94 |
| Mellor Furnace Engr. Co. | 107 |
| Mietz Iron Fdry. & Machine Works, Aug. | 111 |
| Minneapolis Steel & Mach. Co. | 114 |
| Monarch Valve & Mfg. Co. | 96 |
| Morehead Mfg. Co. | 86 |
| Morrin Climax Boiler Co. | 104 |
| Morse Chain Co. | 118 |
| Murphy Iron Works | 101 |
| Nashua Machine Co. | 77 |
| National Pipe Bending Co. | 80 |
| National Tube Co. | 92 |
| Naugatuck Mfg. Co. | 80 |
| Neemes Bros. | 100 |
| Nelson Valve Co. | 97 |
| New Bedford Valve Mfg. Co. | 96 |
| New Process Raw Hide Co. | 90 |
| New York & New Jersey Lubricant Co. | 83 |
| New York Belting & Packing Co., Ltd. | 12 |
| New York Edison Co. | 81 |
| Nicholson File Co. | 93 |
| Northern Equipment Co. | 102 |
| Ohio Blower Co. | 103 |
| Oil City Boiler Works | 105 |
| Oil Well Supply Co. | 105 |
| Open Coil Heater & Purifier Co. | 88 |
| Parson Mfg. Co. | 101 |
| Patterson & Co., Frank L. | 76 |
| Peerless Rubber Mfg. Co. | 8 |
| Penberthy Injector Co. | 13 |
| Pennsylvania Coal & Coke Co. | 98 |
| Pennsylvania Flexible Metallic Tubing Co. | 81 |
| Perfection Grate Co. | 100 |
| Perkins Co., Jno. B. | 96 |
| Pierce Co., Wm. B. | 2d cover |
| Pittsburgh Valve, Fdry. and Construction Co. | 96 |
| Power Specialty Co. | 98 |
| Precision Instrument Co. | 108 |
| Providence Engineering Wks. | 114 |
| Quimby Co., Wm. E. | 89 |
| Reliance Gauge Column Co. | 87 |
| Remington Oil Engine Co. | 112 |
| Richardson-Phenix Co. | 84 |
| Ridgway Dynamo and Engine Co. | 115 |
| Robb Engineering Co., Ltd. | 105 |
| Robertson & Sons, Jas. L. | 75 |
| Rochester Railway & Light Co. | 76 |
| Rogers Co., H. A. | 82 |
| Rollins Engine Co. | 117 |
| Rooksby & Co., E. J. | 112 |
| "S-C" Regulator Co., The | 107 |
| St. John, G. C. | 92 |
| Salamander Grate Bar Co. | 101 |
| Scaife & Sons Co., Wm. B. | 107 |
| Schade Valve Mfg. Co. | 96 |
| Schieren Co., Chas. A. | 111 |
| Schuchardt & Schutte | 92 |
| Schutte & Koerting Co. | 93 |
| Simonds & Co., G. L. | 107 |
| Skinner Engine Co. | 114 |
| Smith, Rudy & Co. | 108 |
| Smooth-On Mfg. Co. | 6 |
| Squires Co., C. E. | 87 |
| Standard Grate Co. | 101 |
| Star Brass Mfg. Co. | 91 |
| Starrett Co., L. S. | 82 |
| Struthers-Wells Co. | 113 |
| Sturtevant Co., B. F. | 112 |
| Terry Steam Turbine Co. | 80 |
| Thermoid Rubber Co. | 11 |
| Toledo Pipe Threading Machine Co. | 96 |
| Treadwell Co., M. H. | 100 |
| Trimont Mfg. Co. | 92 |
| Turbo-Blower Co. | 101 |
| Turner-Fricke Mfg. Co. | 113 |
| Uehling Instrument Co. | 108 |
| Under-Feed Stoker Company of America | 82 |
| Underwood & Co., H. B. | 111 |
| Union Steam Pump Co. | 89 |
| Universal Lubricator Co. | 15 |
| Vogt Machine Co., Henry | 104 |
| Von Kokeritz & Co., R. G. | 98 |
| Vulcan Soot Cleaner Co. | 107 |
| Walton Co., F. S. | 16 and 17 |
| Washburn & Granger | 100 |
| Watson & McDaniel Co. | 89 |
| Watts-Campbell Co. | 115 |
| Westinghouse Machine Co. | 118 |
| Wheeler Mfg. Co., C. H. | 88 |
| Whitlock Coil Pipe Co. | 81 |
| Wickes Boiler Co. | 104 |
| Willcox Engineering Co. | 102 |
| Williams & Co., J. H. | 76 |
| Wing Mfg. Co., L. J. | 100 |
| Wise & Bailey | 98 |
| Worthington, Henry R. | 89 |
| Wright Mfg. Co. | 86 |
| Wyckoff & Son Co., A. | 90 |
| Yale & Towne Mfg. Co. | 92 |
| Yarnall-Waring Co. | 95 |
| Ziermore Regulator Co. | 102 |




# Keep Your Gauges Accurate

## *With an American Dead Weight Gauge Tester*

Inaccurate gauges cause coal waste—accurate gauges prevent coal waste.

Test your gauges regularly with an American Dead Weight Gauge Tester and you'll be sure that they give you correct pressure information. This is the simplest device of its kind made—strong, compact, dependable.

It has all the advantages of the mercury column in much less space.

Tests by dead weights as shown in the illustration.

It should be in your plant whether you use one or many gauges.

Write for full particulars.

## American Steam Gauge & Valve Mfg. Co.

**Boston, Mass.**

New York, 30 Church St.
Pittsburg, 8076 Jenkins Arcade Bldg.
Atlanta, 525 Candler Bldg.
San Francisco, Monadnock Bldg.
Chicago, 130-132 No. Jefferson St.
Montreal, 444 St. James St.

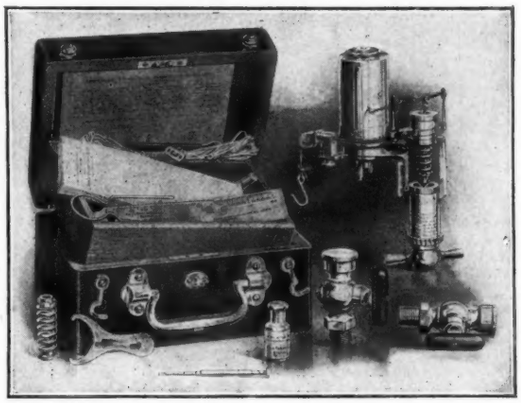
# The American-Thompson Improved Indicator Is The Best Indicator Made

That is the reason we don't meet cheap competition. A cheap indicator is a worthless thing, worse than useless.

A good indicator must be absolutely accurate. That is what the American-Thompson is.

Don't be taken in by the makers of cheap instruments.

If you are going to buy and want to ask others for price, let us give you the names of one or two other high grade instruments; then, compare the instruments and prices. We will depend upon your judgment.



Nothing could be fairer.

Above all, don't buy a cheap indicator, better, none at all.

Finally, bear in mind our original statement:—

The American - Thompson is the best indicator made.

## American Steam Gauge & Valve Mfg. Co.

New York, 30 Church St.
Pittsburg, 8076 Jenkins Arcade Bldg.
Atlanta, 525 Candler Bldg.

**Boston, Mass.**

San Francisco, Monadnock Bldg.
Chicago, 130-132 No. Jefferson St.
Montreal, 444 St. James St.

# CLASSIFIED INDEX *to* ARTICLES ADVERTISED


**Air Compressors**
Allis-Chalmers Co. ..... 83
Ingersoll-Rand Co.....104
Perkins Co., Jno. B..... 96
Union Steam Pump Co..... 89

**Alarms, Low Water**
Elliott Co. ..... 7
Lunkenheimer Co..... 4
Reliance Gauge Column Co... 87
Wright Mfg. Co..... 86

**Alternators**
Allis-Chalmers Co..... 83
General Electric Co..... 78

**Belt Dressing**
Cling-Surface Co. ..... 90
Garlock Packing Co..... 9
Keystone Lubricating Co..... 3
Schieren Co., Chas. A.....111
Smooth-On Mfg. Co..... 6
Walton Co., F. S.....16 and 17

**Belt Lacing**
Bowers Rubber Works..... 99
Bristol Co.....4th cover
Flexible Steel Lacing Co., 4th cover
Morse Chain Co.....118
Schieren Co., Chas. A.....111

**Belting**
Bowers Rubber Works..... 99
Diamond Rubber Co..... 99
Garlock Packing Co..... 9
Morse Chain Co.....118
New York Belting & Pack. Co. 12
Peerless Rubber Mfg. Co..... 8
Schieren Co., Chas. A.....111

**Blocks, Chain**
Morse Chain Co..... 118
Yale & Towne Mfg. Co..... 92

**Blowers, Fan**
Beggs & Co., James..... 84
Buffalo Forge Co..... 92
Ohio Blower Co.....103
Sturtevant Co., B. F.....112
Terry Steam Turbine Co..... 80
Turbo-Blower Co.....101
Wing Mfg. Co., L. J.....100

**Blowers, Steam Jet**
Beggs & Co., James..... 84
McClave-Brooks Co..... 14
Parson Mfg. Co.....101
Schutte & Koerting Co..... 93

**Boiler Compound**
Corliss Chemical Co.....106
Dearborn Drug & Chemical Works ..... 5
Elliott Co. ..... 7
Johns-Manville Co., H. W.... 83
Keystone Lubricating Co..... 3

**Boiler Fronts**
Casey-Hedges Co.....105

**Boiler Setting**
Air-Flex Co.....108
Betson Plastic Fire Brick Co.108
Harbison-Walker Refractories Co. .....109
McLeod & Henry Co.....108

**Boiler Tube Cap Reseating Machine**
Elliott Co. ..... 7
Lagonda Mfg. Co.....3d cover
National Tube Co..... 92

**Boilers**
Allis-Chalmers Co. .....111
Ames Iron Works.....114
Babcock & Wilcox Co.....105
Beggs & Co., James..... 84
Casey-Hedges Co.....105
Griffith & Wedge Co.....114
Griscom-Spencer Co..... 80
Harrisburg Foundry & Mach. Works .....116
Harrison Safety Boiler Wks.. 69
Heine Safety Boiler Co.....104
Hewes & Phillips Iron Wks...116
Holyoke Steam Boiler Wks...107
Keeler Co., E.....104
Kennicott Co..... 81
Kingsford Fdry. & Mach. Co..105
Minneapolis Steel & Mchy. Co.114
Morrin Climax Boiler Co.....104
New Bedford Valve Mfg. Co.. 96
Oil City Boiler Works.....105
Oil Well Supply Co.....105
Robb Engineering Co., Ltd....105
Struthers-Wells Co.....113
Vogt Machine Co., Henry....104
Wickes Boiler Co.....104

**Books**
International Corr. Schools...104
McGraw-Hill Book Co.....110

**Brick, Fire**
Air-Flex Co.....108
Harbison-Walker Refractories Co. .....109
McLeod & Henry Co.....108

**Castings, Brass and Iron**
Bruce-Macbeth Eng. Co.....111
Builders Iron Foundry.....108
Hooven, Owens, Rentschler Co.115
Lunkenheimer Co., The..... 4
McClave-Brooks Co..... 14
Neemes Bros.....109
Salamander Grate Bar Co....101
Treadwell Co., M. H.....109

**Castings, Steel**
Hooven, Owens, Rentschler Co.115

**Cement, Heat Resisting**
Air-Flex Co.....108
Harbison-Walker Refractories Co. .....109

**Cement, Iron**
Johns-Manville Co., H. W.... 83
Smooth-On Mfg. Co..... 6

**Cement, Rubber**
Peerless Rubber Mfg. Co..... 8

**Chemists**
Smith, Rudy & Co..... 108

**Chimneys, Brick**
Kellogg Co., M. W..... 94

**Chimneys, Steel**
Keeler Co., E.....104
Morrin Climax Boiler Co.....104

**Clocks**
American Steam Gauge and Valve Mfg. Co.....66 and 67

**Coal**
Consolidation Coal Co..... 79
Pennsylvania Coal & Coke Co. 98

**Coal and Ash Handling Machinery**
Green Engineering Co.....101
Parson Mfg. Co.....101

**Cocks, Blow-off**
Elliott Co. ..... 7

**Cocks, Gage**
American Steam Gauge and Valve Mfg. Co.....66 and 67
Chaplin-Fulton Co. .....106
Huyette Co., Paul B.....102
Jenkins Bros..... 71
Lunkenheimer Co..... 4
Ohio Blower Co.....103
Reliance Gauge Column Co... 87
"S-C" Regulator Co..... 107
Star Brass Mfg. Co..... 91
Wright Mfg. Co..... 86

**Cocks, Steam**
Dart Mfg. Co., E. M..... 94
Lunkenheimer Co..... 4
National Tube Co..... 92
Schutte & Koerting Co..... 93

**Combustion Chamber, Back Arch**
Air-Flex Co.....108
Betson Plastic Fire Brick Co.108
Harbison-Walker Refractories Co. .....109
McLeod & Henry Co.....108

**Combustion Recorders**
Precision Instrument Co.....108
Uehling Instrument Co.....108

**Compound, Commutator**
Dixon Crucible Co., Jos..... 84

**Condensers**
Alberger Condenser Co..... 89
Allis-Chalmers Co. .....111
Anderson Co., V. D..... 88
Baragwanath & Son, Wm.... 81
Cameron Steam Pump Works, A. S..... 89
Epping-Carpenter Co..... 89
Griscom-Spencer Co..... 80
Kellogg Co., M. W..... 94
Minneapolis Steel & Mchy. Co.114
Schutte & Koerting Co..... 93
Union Steam Pump Co..... 89
Wheeler Mfg. Co., C. H..... 88

**Controllers, Electric**
General Electric Co..... 78

**Controllers, Feed Water**
Chaplin-Fulton Co. .....106
Elliott Co. ..... 7
Lagonda Mfg. Co.....3d cover

**Controllers, Feed Water** —*Continued.*
Mason Regulator Co..... 94
Northern Equipment Co.....102
Squires Co., C. E..... 87
Ziermore Regulator Co.....102

**Cooling Plants**
Schutte & Koerting Co..... 93

**Cooling Towers**
Wheeler Mfg. Co., C. H..... 88

**Cordage**
American Mfg. Co..... 90
Columbian Rope Co..... 90

**Correspondence Schools**
International Corres. Schools. 73

**Couplings, Flexible Shaft**
Bruce-Macbeth Engine Co.....111
Hooven, Owens, Rentschler Co.115

**Covering, Pipe and Boiler**
Johns-Manville Co., H. W.... 83
Wyckoff & Son Co., A..... 90

**Cranes**
Yale & Towne Mfg. Co..... 92

**Crank Pin Oiler**
Richardson-Phenix Co..... 84

**Cylinder Boring Bars**
Rooksby & Co., E. J.....112
Underwood & Co., H. B.....111

**Die Stocks**
Armstrong Mfg. Co..... 94
Bignall & Keeler Mfg. Co.... 94
Borden Co. ..... 94
Curtis & Curtis Co..... 95
Toledo Pipe Threading Machine Co. ..... 96

**Draft Apparatus, Mechanical**
Buffalo Forge Co..... 92
Parson Mfg. Co.....101
Schutte & Koerting Co..... 93
Sturtevant Co., B. F.....112
Turbo-Blower Co.....101
Wing Mfg. Co., L. J.....100

**Drills, Upright**
Barnes Co., W. F. & John....104

**Dust Collectors**
Ohio Blower Co.....103
Sturtevant Co., B. F.....112

**Dynamos**
Allis-Chalmers Co. .....111
American Engine Co.....116
Bruce-Macbeth Eng. Co.....111
General Electric Co..... 78
Ridgway Dynamo and Engine Co. .....115
Terry Steam Turbine Co..... 80

**Economizers, Fuel**
Parson Mfg. Co.....101
Sturtevant Co., B. F..... 112
Westinghouse Machine Co....118

**Ejectors**
Beggs & Co., James..... 84
Lunkenheimer Co..... 4
Penberthy Injector Co..... 13
Schutte & Koerting Co..... 93

**Electric Current**
New York Edison Co..... 81

**Engine Repairs**
Rooksby & Co., E. J.....112
Underwood & Co., H. B.....111

**Engine Stops**
Schutte & Koerting Co..... 93

**Engines, Combustion**
Alberger Gas Engine Co.....113
Allis-Chalmers Co.....111
Bogart Gas Power Engineering Co. .....113
Bruce-Macbeth Eng. Co.....111
Buckeye Engine Co.....115
Du Bois Iron Works.....113
Foos Gas Engine Co.....113
Mietz Iron Fdry. & Mach. Co.111
Minneapolis Steel & Mchy. Co.114
Remington Oil Engine Co....112
Struthers-Wells Co.....113
Turner-Fricke Mfg. Co.....113
Westinghouse Machine Co.... 118

**Engines, Oil**
Foos Gas Engine Co.....113
Mietz Iron Fdry. & Mach. Co.111
Remington Oil Engine Co....112

**Engines, Steam**
Allis-Chalmers Co.....111
American Engine Co.....116
Ames Iron Works..... 114
Ball & Wood Co.....116
Ball Engine Co.....116
Beggs & Co., James..... 84
Buffalo Forge Co..... 92
Buckeye Engine Co.....115
Cooper Co., C. & G.....113
Du Bois Iron Works.....113
Fitchburg Steam Engine Co..115
Griffith & Wedge Co.....114
Griscom-Spencer Co..... 80
Harris Steam Engine Co..116
Harrisburg Fdry & Mach. Wks.116
Hewes & Phillips Iron Wks..116
Hooven, Owens, Rentschler Co.115
Ide & Son, A. L.....116
McIntosh, Seymour & Co....116
Minneapolis Steel & Mchy. Co.114
Perkins, Jno. B..... 96
Providence Engr. Works.....114
Ridgway Dynamo & Engine Co.115
Rollins Engine Co.....117
Skinner Engine Co.....114
Sturtevant Co., B. F.....112
Watts-Campbell Co.....115
Westinghouse Machine Co....118

**Exhaust Heads**
Burt Mfg. Co..... 18
Elliott Co. ..... 7
Hoppes Mfg. Co.....4th cover
Ohio Blower Co.....103
Patterson & Co., Frank I..... 76
Robertson & Sons, Jas. L..... 75
Sturtevant Co., B. F.....112
Whitlock Coil Pipe Co..... 81
Wright Mfg. Co..... 86

**Expansion Joints**
Alberger Condenser Co..... 89
National Tube Co..... 92

**Fans, Electric**
General Electric Co..... 78

**Fans, Exhaust and Ventilating**
Anderson Co., V. D..... 88
Buffalo Forge Co..... 92
General Electric Co..... 78
Power Specialty Co..... 98
Turbo-Blower Co.....101
Wing Mfg. Co., L. J.....100

**Feeders, Furnace**
Ohio Blower Co.....103

**Files and Rasps**
Nicholson File Co..... 93

**Filters, Oil**
Burt Mfg. Co..... 18
Elliott Co. ..... 7
Huyette Co., Paul B.....102
Richardson-Phenix Co..... 84
Robertson & Sons, Jas. L..... 75
Wright Mfg. Co..... 86

**Filters, Water**
Beggs & Co., James..... 84
Griscom-Spencer Co..... 80
Scaife & Sons Co., Wm. B....107

**Fire Clay**
Air-Flex Co.....108
Harbison-Walker Refractories Co. .....109

**Firebox Blocks**
Air-Flex Co.....108
Betson Plastic Fire Brick Co.108
Harbison-Walker Refractories Co. .....109
McLeod & Henry Co.....108

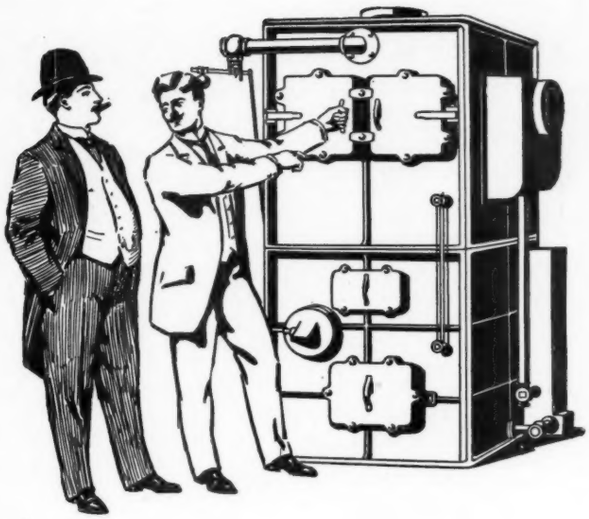
**Fittings, Ammonia**
American Steam Gauge and Valve Mfg. Co.....66 and 67

**Flanges**
Ball & Wood Co.....116
Dart Mfg. Co., E. M..... 94
Jefferson Union Co..... 10
National Tube Co..... 92

**Floats**
Anderson Co., V. D..... 88
Naugatuck Mfg. Co..... 80
Schutte & Koerting Co..... 93

**Flue Gas Analysis Instruments**
Pierce Co., Wm. B.....2d cover

**Furnace Door Arch**
Air-Flex Co.....108
Harbison-Walker Refractories Co. .....109
McLeod & Henry Co.....108



# "EASY TO CLEAN"

is one of the strongest recommendations of the COCHRANE OPEN FEED WATER HEATER. Not only does it refer to the fact that the scale-forming matter can be shoveled out of the sedimentation chamber or scraped off the trays with much less labor than scale can be removed from the boiler, but it also implies that the Cochrane Heater needs cleaning occasionally, that is, it serves to arrest a large amount of impurities that otherwise would reach the boiler.

Pressure heaters, which are virtually closed continuations of the boiler feed pipe, do not, on the other hand, have any noticeable purifying action. They make no provision for the escape of carbonic acid gas, which is necessary for the precipitation of carbonates, and further, the water is kept in rapid circulation (the more rapid the better for the heat transmisson), and as such heaters are not provided with filters, there is little chance of their stopping any sediment. In fact, as far as closed heaters do stop sediment, it is a disadvantage, since it is almost impossible to clean a closed heater, and when closed heaters become clogged up with scale on one side of the heating surface, and with oil and grease on the other, they become practically worthless.

**The full temperature of 210° F., to which the water is raised in the Cochrane Heater,** is not only valuable in saving coal, but brings about the precipitation of carbonates and the escape of corrosive air and gases. If your boiler feed water contains permanent hardness, that is, sulphates, chlorides, or acids, our **Sorge-Cochrane Hot Process System of Water Softening** is recommended. This System takes advantage of the fact that the water should be heated in any case, and that heating will do all or more than lime will do in throwing down the carbonates. The "Hot Process System" combines, in fact, all the valuable qualities of the Cochrane Open Feed Water Heater with those of a perfect softening system, and it costs less, occupies less room and requires less skill and labor to care for and operate, than would the independent heater and softener.

***We would be pleased to advise you in regard to your feed water and heater and softening problems, and to send you our descriptive treatise upon request.***

***Ask for Bulletins "15-H," "15-E" and "15-S."***

COCHRANE SEPARATORS

**Harrison Safety Boiler Works,**

**17th and Clearfield Sts., Philadelphia, Pa.**

COCHRANE HEATERS

**Furnace Linings**
Harbison-Walker Refractories Co. .....109

**Furnaces, Smokeless**
Green Engineering Co. .....101
McClave-Brooks Co. ..... 14
Mellor Furnace Engr. Co. .....107
Murphy Iron Works .....101

**Gage Boards**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Star Brass Mfg. Co. ..... 91

**Gage Glasses**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Chesterton Co., A. W. ..... 88
Garlock Packing Co. ..... 9
Peerless Rubber Mfg. Co. ..... 8
Rogers Co., H. A. ..... 82
Von Kokeritz & Co., R. G. .... 98

**Gage Testing Outfit**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Star Brass Mfg. Co. ..... 91

**Gages, Draft**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Bristol Co. .....4th cover
Star Brass Mfg. Co. ..... 91

**Gages, Pressure**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Ashton Valve Co. ..... 95
Crosby Steam Gage & Valve Co. ..... 95
Star Brass Mfg. Co. ..... 91

**Gages, Recording**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Ashton Valve Co. ..... 95
Bristol Co. .....4th cover
Star Brass Mfg. Co. ..... 91
Uehling Instrument Co. .....108

**Gages, Vacuum**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Schutte & Koerting Co. ..... 93
Star Brass Mfg. Co. ..... 91

**Gages, Water**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Greene, Tweed & Co., 85 and 4th cover
Huyette Co., Paul B. .....102
Jenkins Bros. ..... 71
Lunkenheimer Co. ..... 4
Ohio Blower Co. .....103
Reliance Gauge Column Co. .. 87
Star Brass Mfg. Co. ..... 91
Wright Mfg. Co. ..... 86

**Gas Exhausters**
Ohio Blower Co. .....103

**Gas Producers**
Bogart Gas Power Engineering Co. .....113
Du Bois Iron Works .....113
Foos Gas Engine Co. .....113
Griscom-Spencer Co. ..... 80
Minneapolis Steel & Mchy. Co.114

**Gaskets**
Cancos Mfg. Co. ..... 98
Diamond Rubber Co. ..... 99
Eureka Packing Co. ..... 74
Garlock Packing Co. ..... 9
Greene, Tweed & Co., 85 and 4th cover
Jenkins Bros. ..... 71
Johns-Manville Co., H. W. .... 83
Keeler Co., E. .....104
New York Belting & Packing Co. ..... 12
Peerless Rubber Mfg. Co. ..... 8
Smooth-On Mfg. Co. ..... 6

**Gears**
New Process Raw Hide Co. .. 90

**Generating Sets**
Foos Gas Engine Co. .....113
Mietz Iron Foundry Mach. Co.111

**Governors, Pump**
Chaplin-Fulton Co. .....106
Elliott Co. ..... 7
Mason Regulator Co. ..... 94
Northern Equipment Co. .....102
"S-C" Regulator Co. .....107
Squires Co., C. E. ..... 87
Watson & McDaniel Co. ..... 89

**Governors, Pump Speed**
Mason Regulator Co. ..... 94
Ziermore Regulator Co. .....102

**Graphite**
Dixon Crucible Co., Jos. ..... 84
Garlock Packing Co. ..... 9

**Grates**
Beggs & Co., James. ..... 84
Casey-Hedges Co. .....105
Green Engr. Co. ..... 101
Long Grate Bar Co. .....100
Martin Grate Co. .....100
McClave-Brooks Co. ..... 14
Mellor Furnace Engr. Co. .....107
Neemes Bros. .....100
Perfection Grate Co. .....100
Robertson & Sons, Jas. L. .... 75
Salamander Grate Bar Co. ....101,
Standard Grate Co. .....101
Treadwell Co., M. H. .....100
Washburn-Granger Co. .....100

**Grease**
Albany Lubricating Co..1st cover
Appleby Co., Geo. ..... 85
Dearborn Drug & Chemical Works ..... 5
Dixon Crucible Co., Jos. ..... 84
Keystone Lubricating Co. ..... 3
N. Y. & N. J. Lub. Co. ..... 83
Universal Lubricator Co. ..... 15
Walton Co., F. S. .....16 and 17

**Grease Extractors**
Beggs & Co., James. ..... 84
Elliott Co. ..... 7
Griscom-Spencer Co. ..... 80

**Heaters and Purifiers, Feed Water**
Alberger Condenser Co. ..... 89
Baragwanath & Son, Wm. ..... 81
Casey-Hedges Co. .....105
Elliott Co. ..... 7
Griscom-Spencer Co. ..... 80
Harrison Safety Boiler Wks.. 69
Hoppes Mfg. Co. .....102
Minneapolis Steel & Mchy. Co.114
National Pipe Bending Co. .... 80
Open Coil Heater & Purifier Co. ..... 88
Patterson & Co., Frank L. .... 76
Robertson & Sons, Jas. L. .... 75
Wheeler Mfg. Co., C. H. ..... 88
Whitlock Coil Pipe Co. ..... 81
Wickes Boiler Co. .....104

**Heating and Ventilating Systems**
Ohio Blower Co. .....103
Schutte & Koerting Co. ..... 93
Wing Mfg. Co., L. J. .....100

**Heating, Central Station**
American District Steam Co..111

**Hoists, Combustion Engine**
Foos Gas Engine Co. .....113
Mietz Iron Fdry. & Machine Co. .....111

**Hoists, Electric**
Allis-Chalmers Co. .....111
Yale & Towne Mfg. Co. ..... 92

**Hoists, Hand**
Yale & Towne Mfg. Co. ..... 92

**Hose, Metal**
American Metal Hose Co. .... 82

**Hose, Rubber**
Diamond Rubber Co. ..... 99
Garlock Packing Co. ..... 9
Lagonda Mfg. Co. .....3d cover
Liberty Mfg. Co. ..... 7
New York Belting & Packing Co. ..... 12
Peerless Rubber Mfg. Co. ..... 8
Pennsylvania Flexible Metallic Tubing Co. ..... 81

**Ice and Refrigerating Machinery**
Vogt Machine Co. .....104
Watts-Campbell Co. .....115

**Ice Machine Repairs**
Underwood & Co., H. B. .....111

**Indicators, Ammonia**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Star Brass Mfg. Co. ..... 91

**Indicators, Combustion Engine**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Star Brass Mfg. Co. ..... 91

**Indicators, Flow**
Richardson-Phenix Co. ..... 84

**Indicators, Speed**
Star Brass Mfg. Co. ..... 91
Starrett Co., L. S. ..... 82

**Indicators, Steam Engine**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Crosby Steam Gage & Valve Co. ..... 95
Robertson & Sons, Jas. L. .... 75
Star Brass Mfg. Co. ..... 91

**Injectors**
Jenkins Bros. ..... 71
Lunkenheimer Co. ..... 4
Penberthy Injector Co. ..... 13
Robertson & Sons, Jas. L. .... 75
Schutte & Koerting Co. ..... 93

**Lamps, Electric**
General Electric Co. ..... 78
Johns-Manville Co., H. W. .... 83

**Lathes, Foot Power**
Barnes Co., W. F. & John. ....104

**Lubricators, Cylinder**
Albany Lubricating Co..1st cover
Detroit Lubricator Co. ..... 85
Greene, Tweed & Co., 85 and 4th cover
Griscom-Spencer Co. ..... 80
Lunkenheimer Co. ..... 4
Manzel Bros. Co. ..... 80
Richardson-Phenix Co. ..... 84
Robertson & Sons, Jas. L. .... 75
Star Brass Mfg. Co. ..... 91
Universal Lubricator Co. ..... 15

**Lubricators, Force Feed**
Greene, Tweed & Co., 85 and 4th cover
Lunkenheimer Co. ..... 4
Richardson-Phenix Co. ..... 84
Star Brass Mfg. Co. ..... 91

**Mats and Matting**
Diamond Rubber Co. ..... 99
New York Belting & Packing Co ..... 12
Peerless Rubber Mfg. Co. .... 8

**Metal, Bearing**
Magnolia Metal Co. .....112

**Meters, Steam**
General Electric Co. ..... 78

**Meters, Water**
Builders Iron Foundry .....108
Elliott Co. ..... 7
St. John, G. C. ..... 92

**Micrometers**
Starrett Co., L. S. ..... 82

**Milling Machines, Portable**
Underwood & Co., H. B. .....111

**Motors, Electric**
Allis-Chalmers Co. .....111
American Engine Co. .....116
General Electric Co. ..... 78
Ridgway Dynamo & Engine Co ..... 115
Sturtevant Co., B. F. .....112

**Motors, Water**
Elliott Co. ..... 7
Lagonda Mfg. Co. .....3d cover

**Oil and Grease Cups**
Albany Lubricating Co..1st cover
Detroit Lubricating Co. ..... 85
Elliott Co. ..... 7
Griscom-Spencer Co. ..... 80
Keystone Lubricating Co. .... 3
Lunkenheimer Co. ..... 4
N. Y. & N. J. Lubricant Co. .. 83
Richardson-Phenix Co. ..... 84
Star Brass Mfg. Co. ..... 91
Universal Lubricator Co. ..... 15

**Oil Burners**
Hammel Oil Burner Co. ..... 83
Lunkenheimer Co. ..... 4
Parson Mfg. Co. .....101

**Oil Reservoirs**
Burt Mfg. Co. ..... 18
Richardson-Phenix Co. ..... 84

**Oiling Systems**
Albany Lubricating Co..1st cover
Burt Mfg. Co. ..... 18
Detroit Lubricator Co. ..... 85
Elliott Co. ..... 7
Lunkenheimer Co. ..... 4
Richardson-Phenix Co. ..... 84

**Oils**
Albany Lubricating Co..1st cover
Dearborn Drug & Chemical Works ..... 5
N. Y. & N. J. Lubricant Co. .. 83
Walton Co., F. S. .....16 and 17

**Packing, Flange**
Bowers Rubber Works. ..... 99
Cancos Mfg. Co. ..... 98
Chesterton Co., A. W. ..... 88
Diamond Rubber Co. ..... 99
Eureka Packing Co. ..... 74
Garlock Packing Co. ..... 9
Greene, Tweed & Co., 85 and 4th cover
Jenkins Bros. ..... 71
Johns-Manville Co., H. W. .... 83
New York Belting & Packing Co. ..... 12
Peerless Rubber Mfg. Co. .... 8

**Packing, Flange**—*Continued.*
Smooth-On Mfg. Co. ..... 6
Thermoid Rubber Co. ..... 11
Von Kokeritz & Co., R. G. .... 98
Wise & Bailey. ..... 98

**Packing, Hydraulic**
Cancos Mfg. Co. ..... 98
Chesterton Co., A. W. ..... 88
Diamond Rubber Co. ..... 99
Garlock Packing Co. ..... 9
Johns-Manville Co., H. W. .... 83
Robertson & Sons, Jas. L. .... 75
Schieren Co., Chas. A. .....111
Von Kokeritz & Co., R. G. .... 98

**Packing, Piston Rod**
Bowers Rubber Works. ..... 99
Cancos Mfg. Co. ..... 98
Chesterton Co., A. W. ..... 88
Diamond Rubber Co. ..... 99
Eureka Packing Co. ..... 74
Garlock Packing Co. ..... 9
Greene, Tweed & Co., 85 and 4th cover
Johns-Manville Co., H. W. .... 83
New York Belting and Packing Co. ..... 12
Peerless Rubber Mfg. Co. .... 8
Power Specialty Co. ..... 98
Thermoid Rubber Co. ..... 11
Von Kokeritz & Co., R. G. .... 98
Wise & Bailey. ..... 98

**Packing, Pump Valve**
Cancos Mfg. Co. ..... 98
Diamond Rubber Co. ..... 99
Von Kokeritz & Co., R. G. .... 98
Wise & Bailey. ..... 98

**Packing, Valve Stem**
Cancos Mfg. Co. ..... 98
Diamond Rubber Co. ..... 99
Greene, Tweed & Co., 85 and 4th covtr
Wise & Bailey. ..... 98

**Pipe**
Ball & Wood Co. .....116
Kellogg Co., M. W. ..... 94
National Tube Co. ..... 92

**Pipe Bending**
Ball & Wood Co. .....116
Kellogg Co., M. W. ..... 94
National Pipe Bending Co. .... 80
Whitlock Coil Pipe Co. ..... 81

**Pipe Clamp**
Yarnall-Waring Co. ..... 95

**Pipe Coils**
National Pipe Bending Co. .... 80
National Tube Co. ..... 92
Whitlock Coil Pipe Co. ..... 81

**Pipe Cutting and Threading Machines**
Armstrong Mfg. Co. ..... 94
Bignall & Keeler Mfg. Co. .... 94
Borden Co. ..... 94
Curtis & Curtis Co. ..... 95
Toledo Pipe Threading Mach. Co. ..... 96
Trimont Mfg. Co. ..... 92
Williams & Co., J. H. ..... 76

**Piping, Blower**
Ohio Blower Co. .....103

**Piping, High Pressure**
Kellogg Co., M. W. ..... 94

**Planimeters**
American Steam Gauge and Valve Mfg. Co. .....66 and 67
Robertson & Sons, Jas. L. .... 75

**Plumbago**
Dixon Crucible Co., Jos. ..... 84

**Polish, Metal**
Hoffman, Geo. W. ..... 88
Johns-Manville Co., H. W. .... 83

**Publishers**
International Corres. Schools. 104
McGraw-Hill Book Co. .....110

**Pulleys**
Minneapolis Steel & Mchy. Co.114

**Pumps, Air**
Cameron Steam Pump Wks., A. S. ..... 89
Deming Co. ..... 89
Du Bois Iron Works .....113
Wheeler Mfg. Co., C. H. ..... 88

**Pumps, Centrifugal**
Alberger Condenser Co. ..... 89
Allis-Chalmers Co. .....111
Deming Co. ..... 89
Quimby, Inc., Wm. E. ..... 89
Terry Steam Turbine Co. ..... 80
Worthington, Henry R. ..... 89

**Pumps, Deep Well**
Cameron Steam Pump Wks., A. S. ..... 89

# Feed The Oil Where It's Needed And Save The Hose

The Lagonda Air and Steam Driven Boiler Tube Cleaner is a crackerjack for removing scale. It removes the hardest scale, and leaves the tubes clean as a whistle. But here's an improvement which makes this cleaner better than ever. It is the **Lagonda Automatic Lubricator.** It saves at least 50% of the oil, prolongs the life of the cleaner 50% and increases the power and efficiency.

The **Lagonda Automatic Lubricator** consists of a sight feed oil cup located in the steam or air line outside of the boiler. The rate of feeding oil is always under the control and supervision of the operator. **The main thing is that the oil is fed through a flexible metallic hose, INSIDE of the main hose. The oil is delivered directly to the cleaner, none ever coming in contact with the rubber hose.** You know oil on rubber soon puts it out of business.

WEINLAND AIR OR STEAM DRIVEN CLEANER

**This lubricator can also be used for all classes of air driven tools such as air riveters, hammers, chisels, etc.**

**Get our new catalog on this subject and get posted on up-to-date lubrication methods.**

## The Lagonda Mfg. Co.

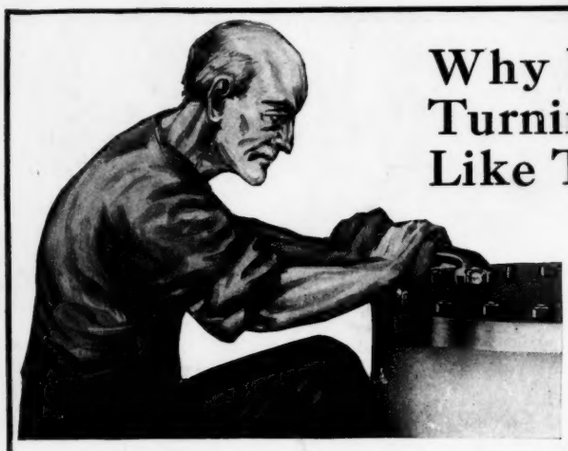
SPRINGFIELD, OHIO.

FUEL SAVING SPECIALISTS

NEW YORK, BOSTON, PHILADELPHIA, PITTSBURG, CHICAGO, DETROIT, ST. LOUIS SAN FRANCISCO ST. PAUL MONTREAL, LONDON

*Makers of Weinland Boiler Tube Cleaners, Automatic Cut-Off Valves, Reseating Machines, Boiler Tube Cutters and Strainers.*

161



# Why Waste Time, Turning Nuts With The Old Fashioned Wrench Like This?

When by using the

## Favorite Reversible Ratchet Wrench

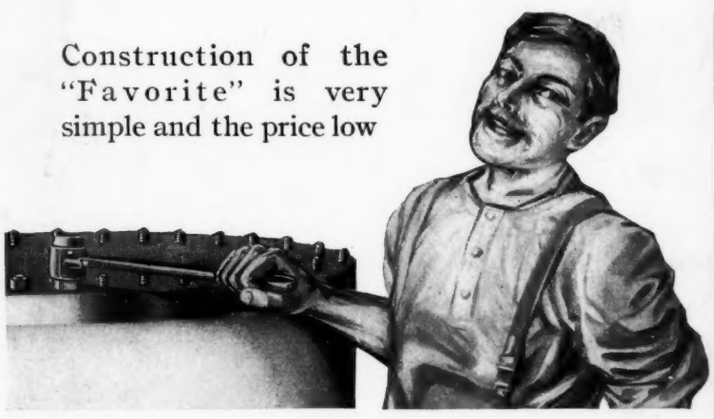
you can get the nuts on or off quickly without damage.

You save time because of the ratchet movement of the wrench, which is continuous.

No half turn and a fresh hold, which often means skinned knuckles and burred nuts.

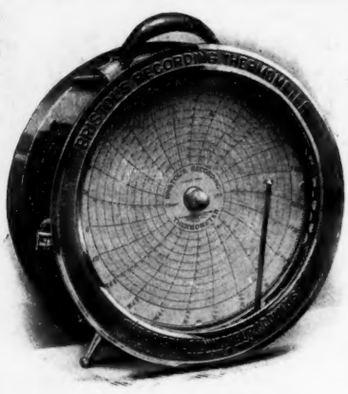
If you only knew it, you cannot afford to be without the "Favorite" in the engine room.

Construction of the "Favorite" is very simple and the price low



**Greene, Tweed & Co.**
Sole Manufacturers
**109 Duane St.** **New York**

---



## BRISTOL'S Recording Instruments For Pressure Temperature and Electricity

Unique in their extreme simplicity of construction.

Most complete line of Recording Instruments in the world.

Bristol's Portable Class I Self-Contained Recording Thermometer. The above illustration shows a high quality Instrument of moderate price for recording atmospheric temperatures at many different points.

Illustrated Bulletin No. 125 describing Bristol's Class I Recording Thermometers and other catalogues of Bristol's Instruments will be mailed upon request.

**The Bristol Company** Waterbury, Conn.
BRANCH OFFICES: New York, Pittsburg, Chicago

---

"A rubber belt 8″ X 8-ply was fastened together with a No. 65 Alligator lacing. And both belt and lacing held fast up to a pull of 11, 800 lbs. when the lacing was torn out of the belt."

Send today for FREE Sample.

## Alligator Flexible Steel Belt Lacing

is making records like this right along—records that put it in a class by itself for strength and all-round superiority.

This particular record was made in a test recently held by the Mechanical Engineer at The Armour Institute of Technology, Chicago, in different kinds of belting fastened with both Alligator and other types of belt lacing.

Another record made was this: A 4″ x 4-ply stitched canvas belt with a wire lacing was put under a pull of 1740 lbs. The wire cut clean through the belt. The same belt laced with Alligator withstood a strain of 3070 lbs. before parting.

Not only does "Alligator" hold in this remarkable manner, but no tool is needed in applying, but a hammer.

The thing to do is to test "Alligator" for yourself at our expense. You can do it by sending for Free Sample.

**Flexible Steel Lacing Company**
**542 West Jackson Boulevard** **Chicago, Ill.**
EUROPEAN AGENTS—J. B. Stone & Co., 135 Finsbury Pavement, London, England